FINAL REPORT

Demonstration of an Energy-Efficient Secondary Loop HFC-152a Mobile Air Conditioning System

By

Mahmoud Ghodbane
Timothy D. Craig
James A. Baker

Delphi Thermal Systems Division
Delphi Corporation

for

U.S. Environmental Protection Agency

c/o Kristen Taddonio, Project Officer
Strategic Climate Projects
Climate Protection Partnership Division
1200 Pennsylvania Avenue, MC 6202 J
Washington, D.C. 20460

Order No. EP07H001055

December 17, 2007

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I. INTODUCTION

All currently manufactured automobile and light duty truck vehicle air conditioning systems use HFC-134a as the refrigerant. HFC-134a replaced CFC-12 in the early 1990's due to the negative impact of CFC's on the Earth's ozone layer. While other refrigerants were studied at that time as possible replacements, HFC-134a was chosen for its combination of refrigerating properties and safety characteristics, despite indications that its global warming potential (GWP) and resulting climate impact could one day become a concern. That concern was realized with the recent passage of legislation in the European Community that requires the use of refrigerants with global warming potentials less than 150 in all new-type vehicles starting in 2011 and in all new vehicles by 2017.

The chemical industry has been working to synthesize new chemicals and mixtures of chemicals that might one day prove useful, but has met with limited success to date. Only two commercially available refrigerants exist today that have a GWP below the required threshold, these being carbon dioxide (R-744) and HFC-152a (R-152a). Recently, DuPont and Honeywell have proposed HFO1234yf as R-134a replacement. The automotive industry will likely have to choose between these alternatives very soon to meet the EU timeline for implementation.

R-744 is a refrigerant that requires completely new components, engineered to withstand its inherently very high system pressures, which results in high capital expenditures and unique system development challenges, especially to limit refrigerant leakage. In addition, safety concerns associated with the health effects of excessive leakage into the passenger compartment will likely require engineering mitigation, at least in the United States [1].

R-152a is very similar to the current R-134a in terms of pressure, refrigerating characteristics, and compatibility with currently used system components and materials, making it easier and less expensive to implement. It does, however, suffer from the disadvantage of being mildly flammable, which can be successfully mitigated by applying the refrigerant to a secondary loop system, in which the refrigerant is contained completely within the engine compartment where it chills a secondary fluid (similar to engine coolant used for heating) that is then pumped into the passenger compartment to cool cabin air [1]. This requires additional components, but is still a lower cost and lower commercial risk option than R-744. As far as HFO1234yf is concerned, the jury is still out on its candidacy as R-134a replacement.

The US Environmental Protection Agency has recognized the environmental opportunities R-152a can offer to the vehicle air conditioning industry and the importance of demonstrating this technology in a vehicle system that highlights energy efficiency and commercially viable components. To these ends, this demonstration program was created.

II. BACKGROUND

Delphi has a long-standing history of secondary loop systems development, beginning with the writing of an internal document on the use of flammable refrigerants in 1997 [2]. This led to constructing a secondary loop bench test system in 1998 for hands-on system development. Following successful laboratory testing, vehicle systems were built, tested and demonstrated publicly. The idea was to highlight the potential for using a secondary loop system to enable the use of a mildly toxic or mildly flammable refrigerant as a means of satisfying potential legislation requiring the use of low GWP refrigerants [3].

The first demonstration was a proof-of-concept exercise using identical 1999 Chevrolet Malibu's, one equipped with a secondary loop R-134a system and the other a regular production, direct expansion R-134a system. These vehicles were demonstrated at the 1999 Phoenix Forum (currently the SAE Alternate Refrigerant Systems Symposium) during which participants were invited to ride in both vehicles and see if they could identify the secondary loop vehicle. Although it had been postulated that cooling rate of the secondary loop system would lag behind that of the production system due to the intermediate coolant, the participant ratings showed equivalent cooling performance and comfort. Whereas a temperature lag is measurable during cool down, it is not perceptible by the vehicle occupants. This is due to the rapid temperature change associated with the cooling down of a hot vehicle and the fact that the bulk of the coolant is stored under-hood where its temperature remains near ambient, unlike the passenger cabin, which receives significant additional heating from solar radiation. This demonstration followed the publication of an SAE paper on the use of flammable refrigerants [4].

At the 2003 SAE Alternate Refrigerant System Symposium, Delphi and Volvo jointly demonstrated a Volvo XC90 (SUV) R-152a secondary loop system with proof-of-concept (pre-prototype) components [5].ref. This testing was quite successful with the cooling performance of the XC90 being among the best of the vehicles tested.

At the 2007 SAE Alternate Refrigerant System Symposium, Delphi demonstrated several potential alternatives in direct expansion systems being proposed by the chemical companies. As part of the current project with the EPA, Delphi also successfully demonstrated an R-152a secondary loop system, built and tested by Delphi with funding provided by the US EPA [6]. All components were commercially viable in a system designed to deliver energy efficient operation.

In addition to the activities noted above, Delphi engineers have published numerous SAE technical papers on the topics of alternative refrigerants and A/C system energy efficiency [7]. These are cited in the reference section of this report. Secondary Loop R-152a Demonstration Vehicle Project Overview (annotated original submission to EPA)

III. EPA PROJECT

The Project Plan, broadly described, consists of measuring the cooling and energy performance of an A/C system on a donated 2007 Opel Astra. This is followed by retrofitting that same vehicle with a commercially viable secondary loop HFC-152a system. The vehicle was retested for cooling performance and energy, and then demonstrated the vehicle with the secondary loop system at the 2007 SAE Alternate Refrigerant Systems Symposium at their mid-July meeting in Scottsdale, Arizona.

3.1 Project Objective

System design consists of fully understanding the current physical and operational characteristics of the Astra and applying that knowledge to designing the necessary new components to create the desired secondary loop system. Design details and rationale will be provided to a Peer Review Team for input and concurrence. At this time, the Peer Review Team consists of William Hill (GM), Hans Fernqvist (Volvo), Stephen Andersen (EPA), and Ward Atkinson (SAE). Additional globally recognized experts (e.g., R. Monforte (Fiat) and J. Rugh (NREL) will be invited to join. A Peer review meeting (in person or via teleconference) will be held to discuss initial component and system designs, as well as the intended test methodologies to quantitatively measure system characteristics. A second meeting(s) will be held during the testing phase for those interested in viewing the data of the secondary loop system. Outputs from these meetings that would contribute to improved system performance will be incorporated into the design guide.

Attached in appendix A: Power point file (EPA_R152a_Project_overview.ppt) containing diagrams that both reference, and further describe, the new components to be designed, developed and incorporated into the system. In appendix B: The first Peer Review Team Meeting Presentation. The link to the presentation given at 2007 SAE Alternate Refrigerant Symposium given in Scottsdale, AZ is below: http://www.sae.org/events/aars/presentations/2007aarsvehicles.pdf

3.2 Testing and Demonstration

Also included in the attached PowerPoint file is the summary of the testing protocol for the two system architectures. These are, of course, identical except for the necessary determination of proper system charge for the secondary loop system prior to testing.

3.3 Timing

DELPHI

Thermal R&D

Project Plan

Project Title: Secondary Loop R-152a Demonstration Vehicle
Project Leader: Mahmoud Ghodbane
Technical Team: James Baker, Shawn Caple, Donald Enzinna, William Kumpf, Xiaoxia Mu, David Polisoto, Paul Droman, Timothy Craig, Lindsey Leitzel

Project Number(s): EP07H001055, EPA PO# (CL) (CL) (CL)

Approval Date: 3/1/2007

Revision Date: 11/9/2007

Program Event			Start		Finish		Status	Comments
			Planned (mm/dd/yy)	Actual (mm/dd/yy)	Planned	Actual	✓	
1	Preliminary System Design and Testing Plan		02/01/07	02/01/07	02/21/07	02/26/07	✓	
2	Sub-System Modeling and Optimization		02/15/07	02/15/07	03/09/07	04/20/07	✓	Extended Optimization, No IHX
3	Bench Test of Preliminary Design Secondary Loop Sub-System (SLSS)		03/12/07	03/12/07	03/16/07	04/20/07	✓	Matched to Optimization; Pump & Heat Ex.
4	Vehicle Packaging of SLSS and Components		02/26/07	02/26/07	03/09/07	05/11/07	✓	CAD study supported by vehicle build
5	Peer Design Review #1		03/19/07	04/16/07	03/19/07	04/16/07	✓	At SAE Congress; component data avaiable
6	Select Final SL Design		03/20/07	03/20/07	04/20/07	04/20/07	✓	
7	Peer Design Review # 2		--		--			Precluded based on accelerated timing
8	Demonstration Vehicle -- Tunnel Testing							
	8a	Ship Vehicle to Lockport, NY Technical Center (LTC)	03/19/07	03/19/07	03/30/07	04/23/07	✓	Start date for Vehicle work
	8b	Baseline Test Vehicle with R-134a System	04/09/07	04/23/07	04/13/07	05/04/07	✓	
	8c	Update Vehicle with SLSS	04/16/07	05/14/07	04/20/07	05/31/07	✓	
	8d	Test Vehicle with SLSS	04/23/07	06/01/07	04/27/07	06/08/07	✓	
	8e	Update SLSS Based on 8d tests	05/07/07	06/12/07	05/11/07	06/15/07	✓	Instrument for road testing
	8f	Retest Vehicle with SLSS Updates	05/16/07	--	05/18/07	--		Not necessary
	8g	Validate Vehicle Performance In Tunnel	05/16/07	06/18/07	05/18/07	06/22/07	✓	Controls calibration
9	Peer Performance Review		05/24/07	04/12/07	05/24/07	07/13/07	✓	Multiple reviews with GM on data.
10	Demonstration Vehicle -- Road Testing							
	10a	Prepare Vehicle for Road Test and Data Collection	05/28/07	06/12/07	05/30/07	06/15/07	✓	
	10b	Ship Vehicle to Phoenix, AZ	06/04/07	06/22/07	06/08/07	07/12/07	✓	
	10c	Vehicle Road Tests	06/19/07	07/17/07	06/22/07	07/19/07	✓	SAE ARSS Evaluations
11	Design Manual							
9a		Initial Draft	08/21/07	07/02/07	08/21/07	07/17/07	✓	Review and summary at SAE ARSS
9b		Interim Draft	09/15/07	--	09/15/07	--		
9c		Final Manual	09/28/07	10/29/07	09/28/07	11/28/07	✓	Adjusted target based on vehicle usage

3.4 Design Guidelines

As noted above, design guidelines to maximize secondary loop R-152a system energy efficiency will be collated into design guidelines to accompany the final report.

As requested in the statement of work, EPA will have all property rights and ownership of Design Manual. It is understood that property rights and ownership of the Design Manual means the manual itself, as a tangible publication, and not ownership of the intangible and potentially patentable ideas disclosed therein.

IV. PRODUCTION R-134a VEHICLE AND SYSTEM DESCRIPTION

A 2006 Opel Astra production vehicle was used in the investigation. The actual vehicle is shown in figure 1. It is equipped with a manually controlled air conditioning system.

Figure 1 2006 Opel Astra *(1.3 liter gasoline engine)*

The production R-134a system is laid out schematically in figure 2. The system is comprised of an R-134a refrigerant circuit consisting of components such as compressor, condenser, expansion device, and refrigerant lines located under-hood and an evaporator positioned inside the HVAC case, which is situated in the passenger compartment of the vehicle.

V. R-152a SECONDARY LOOP SYSTEM DESCRIPTION

It was shown [2] that R-152a (1,1-Difluoroethane) is an excellent refrigerant. Obviously, the only negative characteristic of R-152a is its mild flammability. One way to overcome the safety aspect of the mildly flammable refrigerant is to use it in conjunction with a secondary loop cooling system. Figure 3 depicts a solid model of a Secondary loop system.

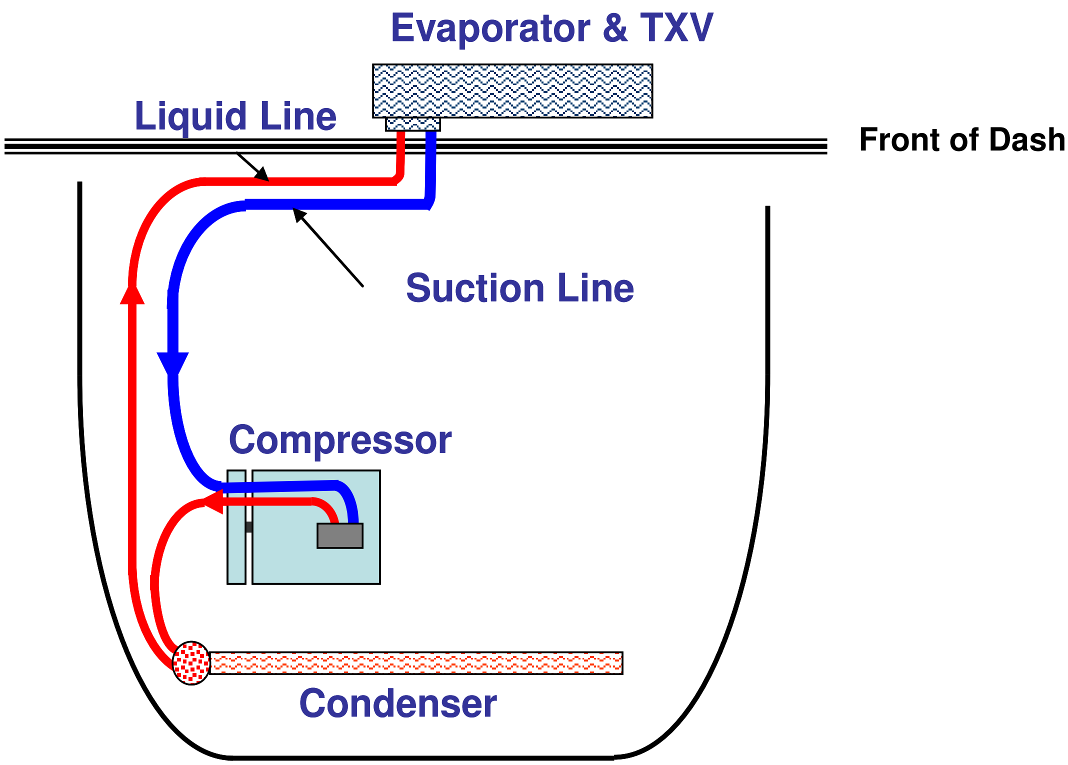

Figure 2 Opel Astra R-134a A/C System Layout
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The conventional evaporator is replaced by a secondary fluid (coolant) to air heat exchanger called a cooler. The secondary loop system uses an R-152a refrigerant direct expansion cooling system (primary system) to cool the coolant via a refrigerant to coolant heat exchanger called a chiller. The cold coolant is then pumped trough the cooler inside the passenger compartment of the vehicle to provide the necessary cooling.

In order for a Secondary loop system to be viable alternative, the heat transfer resistance resulting from the added level of heat exchange must be minimized through proper selection of the coolant and system design. In addition, the packaging and mass of the pump, reservoir, and chiller must be optimized. Unlike the primary refrigerant, it is necessary for the secondary fluid to be non

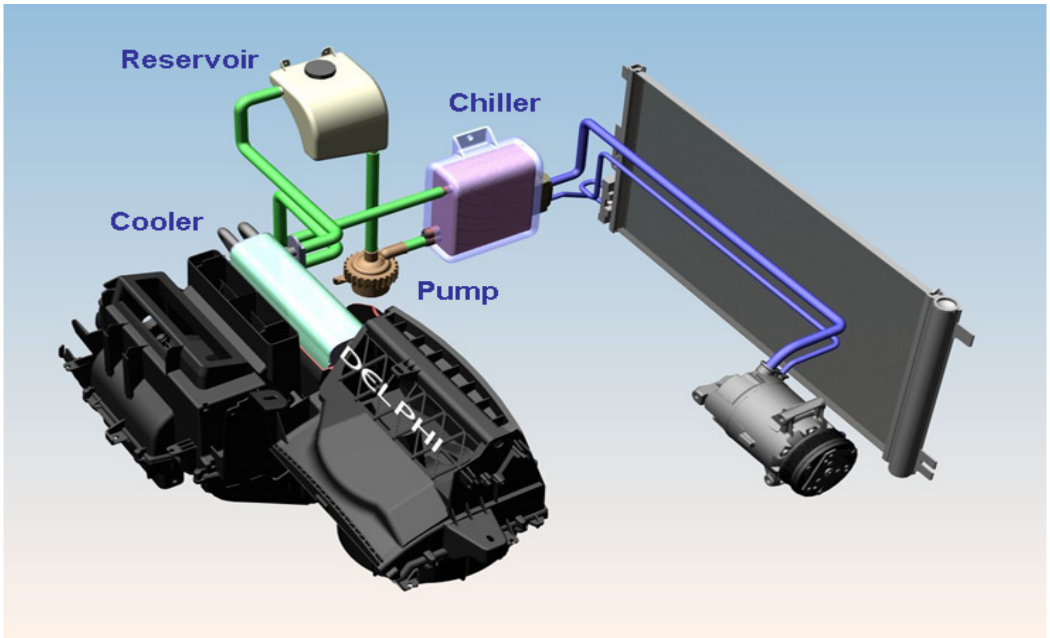

Figure 3 Solid Model of Secondary loop system

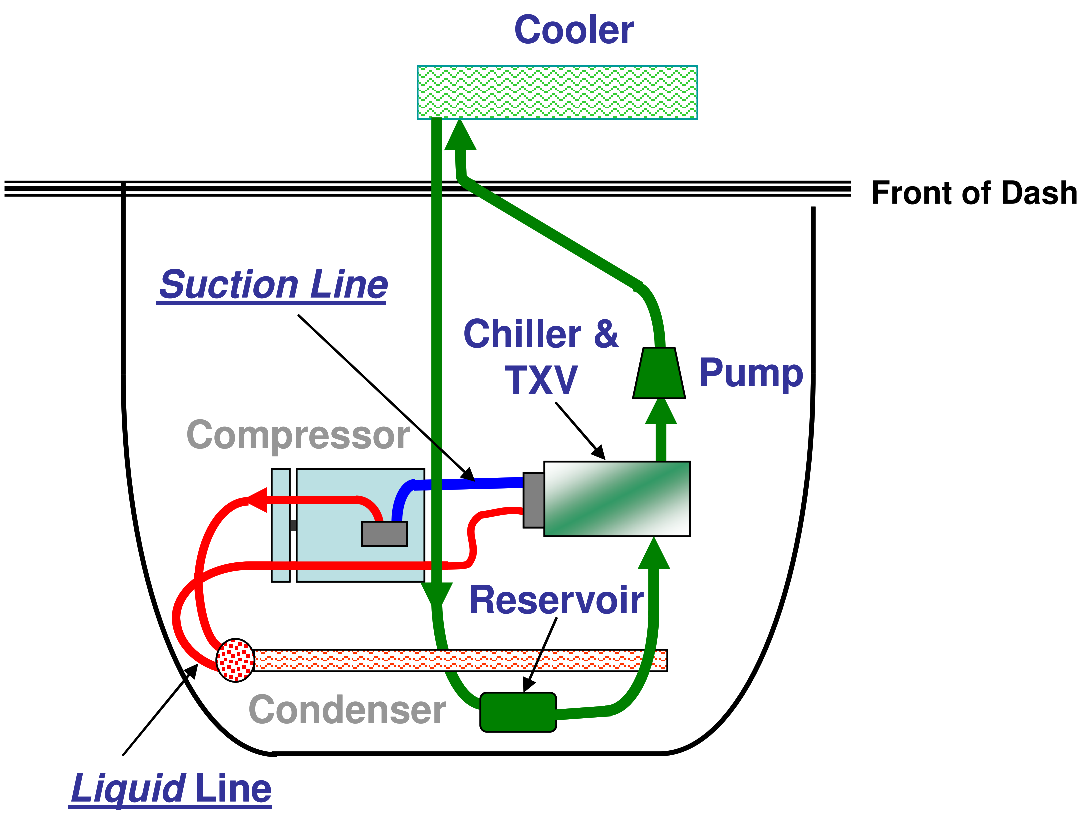

Figure 4 Opel Astra R-152a Secondary Loop System Layout

flammable and non-toxic because it is circulated inside the passenger compartment of the vehicle. There are a variety of commercially developed single-phase heat transfer fluids, which could be used as coolants. In this investigation, ethylene glycol / water solution was chosen because it is common, wildly used, and validated for automotive applications. The selection of the percentage of the ethylene glycol used in the coolant is detailed later in the report.

Figure 4 shows in a schematic fashion the lay out of the Opel Astra Secondary loop system. The system encompasses a primary refrigerant circuit consisting of compressor, condenser, and an expansion device coupled with the coolant circuit through the chiller. In this configuration, the chiller is located up front, near the compressor and cools the circulating coolant. The amount of coolant flow to the cooler inside the vehicle is regulated by a variable speed pump. When comparing the baseline lay out shown in figure 2 and the secondary loop system lay out in figure 4, the primary refrigerant suction and liquid lines of the secondary loop system are much shorter than the baseline system. The short suction line introduces low refrigerant pressure drop, which translates into performance improvement.

VI. SECONDARY LOOP SYSTEM COMPONENTS

As shown previously in figure 3 the secondary loop system is comprised of primary refrigerant circuit consisting of compressor, condenser, and expansion device coupled with a secondary fluid circuit via a chiller. The secondary fluid circuit is composed of a reservoir, a pump, and a cooler taking place of the evaporator inside the HVAC module. The chiller, which is thermally equivalent to an evaporator, is the link between the refrigerant circuit and the secondary fluid circuit.

6.1 Compressor

As in the direct system, the compressor still is the heart of the secondary loop system, and consumes most of the energy. The sizing and the selection of the compressor is critical to any system and in the case of secondary loop system, it is more significant because of the added thermal resistance introduced by the secondary fluid.

If equivalent cooling capacity and similar compressor technology is used for the baseline and the secondary loop system, the displacement of the secondary loop compressor is expected to be higher (10 to 15%) than the baseline compressor; however, this increase can be reduced to a single digit if design and efficiency improvements of the compressor are realized. A theoretical evaluation of the displacement increase can be expressed by the following equation:

$$\left(\frac{Disp_{SL-R152a}}{Disp_{R134a}}\right)=\left(\frac{\rho_{suct-R134a}}{\rho_{suct-R152a}}\right)\left(\frac{\Delta h_{evap-R134a}}{\Delta h_{evap-R152a}}\right)\left(\frac{\eta_{vol-R134a}}{\eta_{vol-R152a}}\right)\left(\frac{RPM_{R134a}}{RPM_{R152a}}\right)$$

Where D_{isp} is the compressor displacement. ρ , Δh, η, and RPM represent refrigerant density, enthalpy difference across the compressor, isentropic efficiency of the compressor, and compressor speed respectively. Using the saturation properties of the refrigerants at 41(5), 50(10), 59 15), and 81 ^{0}F (27 ^{0}C) compressor in temperatures with the assumption that both compressors have identical speed and isentropic efficiency, on the average the R-152a compressor displacement is higher by 11.5%.

6.2 Condenser

Identical condenser and fan were used for the baseline and the secondary loop system. As in any mobile air conditioning, front air management is important for optimum system performance especially during engine idling situations. When the vehicle is idling, re-circulation of the warm air raises the average air temperature entering the condenser and therefore reduces the capacity of the condenser. The re-circulation of warm air may be also caused by insufficient clearance between the engine and the front-end module in which case the warm air from the front-end module is partially reflected back to the condenser. An adequate shrouding of condenser, fan, and radiator module was employed to limit warm air re-circulation. Sufficient shrouding of the front-end module can prevent re-circulation of the warm air discharged from the module.

6.3 Chiller

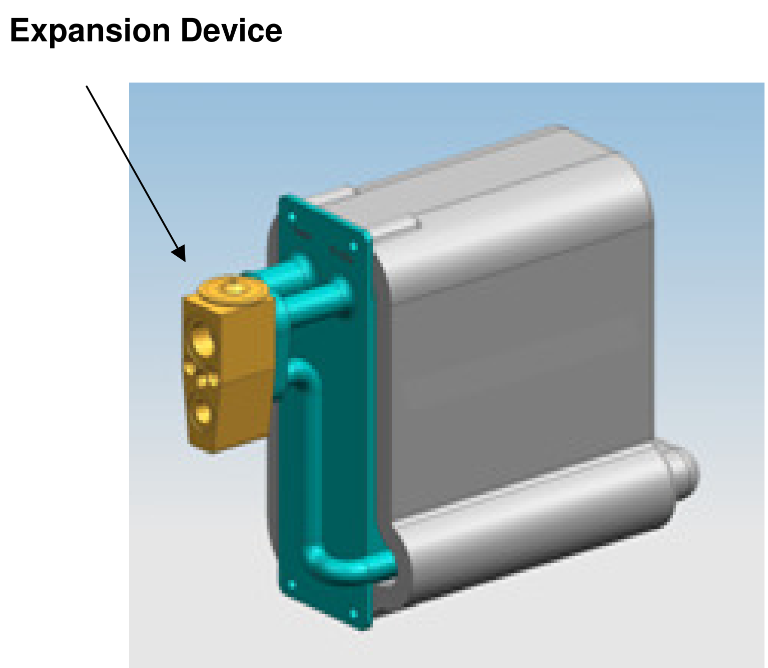

Figure 5 Typical Chiller

The chiller is a refrigerant to secondary fluid (coolant) heat exchanger. In a secondary loop system, the refrigerant evaporation occurs in the chiller instead of the evaporator for a baseline system. Technically the chiller is an evaporator using the coolant as heat transfer medium instead of the air. The chiller used in this investigation is shown in figure 5.

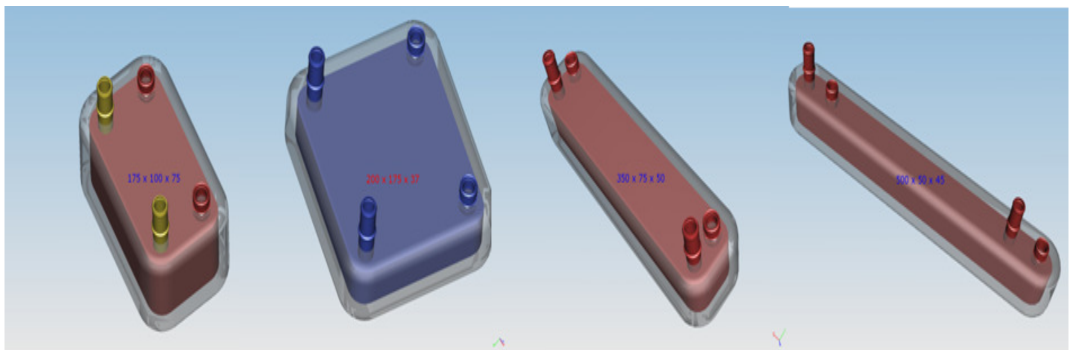

Figure 6 Chiller Options

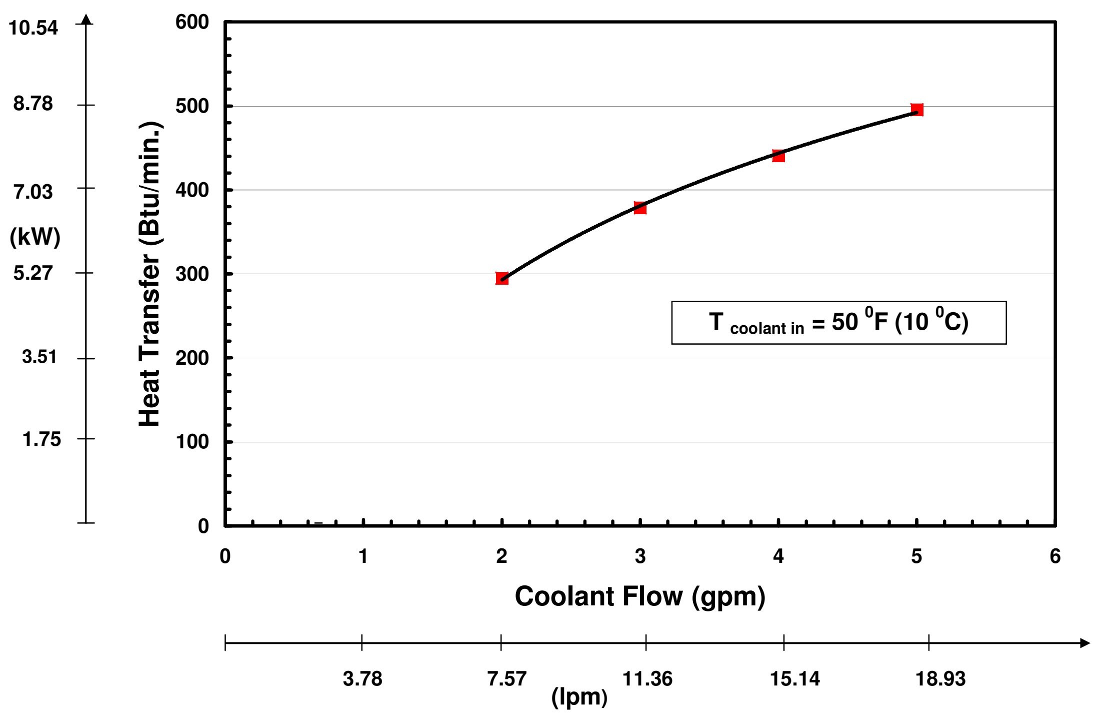

Figure 7 Chiller Heat Transfer Performance

Currently at the prototype stage, the chiller is a brazed aluminum plate type with enhanced heat transfer surfaces for optimum heat transfer and pressure drops. Different shapes and designs are portrayed in figure 6.

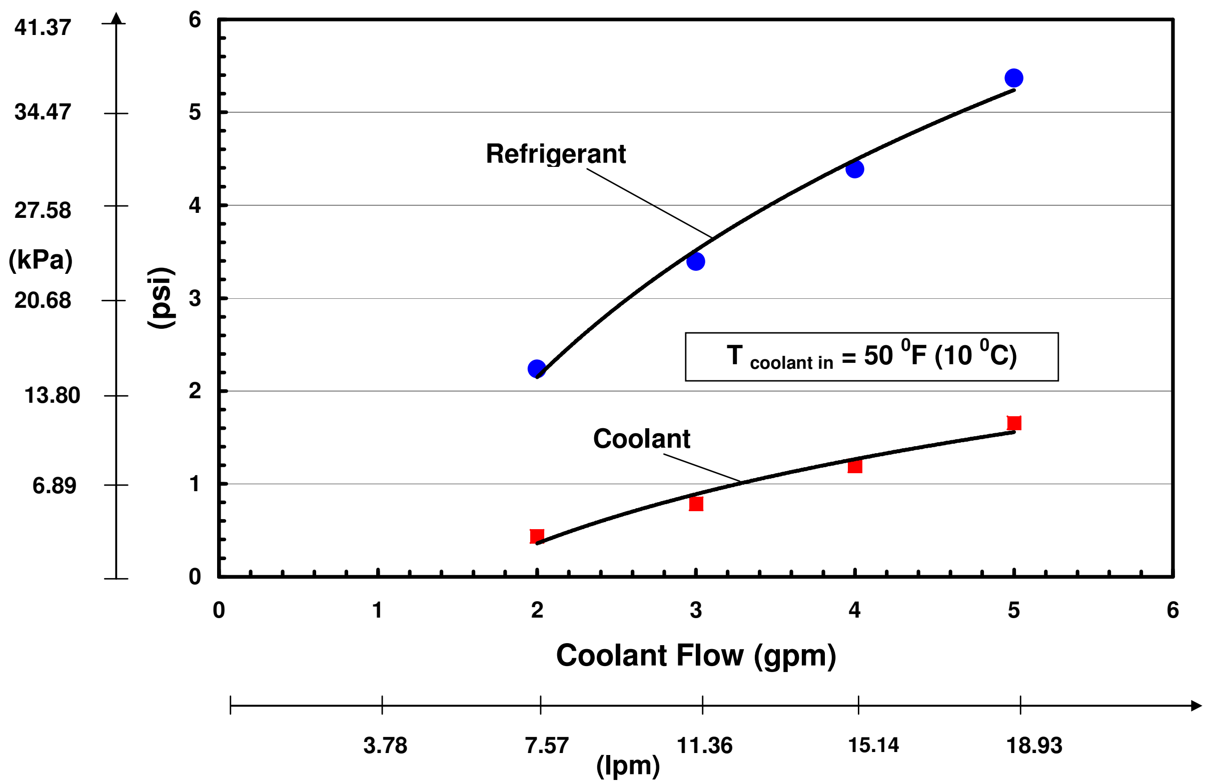

Figure 8 Chiller Pressure Drop

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Figures 7 and 8 depict the thermal performance of the chiller used in this investigation. Figure 7 includes the heat transfer at fixed coolant in temperature and different coolant flow rates. Figure 8 shows the variation of the refrigerant and the coolant pressure drops with the coolant flow rate at fixed chiller coolant in temperature. As expected, the heat transfer increases with coolant flow rate. The data has also shown that the heat transfer drops off when the chiller coolant in temperature falls. Similar curves to figures 7 and 8 can be generated for different chiller coolant in temperatures.

6.4 Refrigerant Controls

The same type and construction of expansion devices as the baseline, fixed orifice or a thermal expansion valve can be used in a secondary loop system. The device must be adjusted to reflect the R-152a refrigerant properties. In most cases, the refrigerant chiller out superheat setting for the secondary loop system is lower than the evaporator out superheat for the baseline system. As shown in figure 9, the low chiller superheat enables the chiller to operate at higher effectiveness.

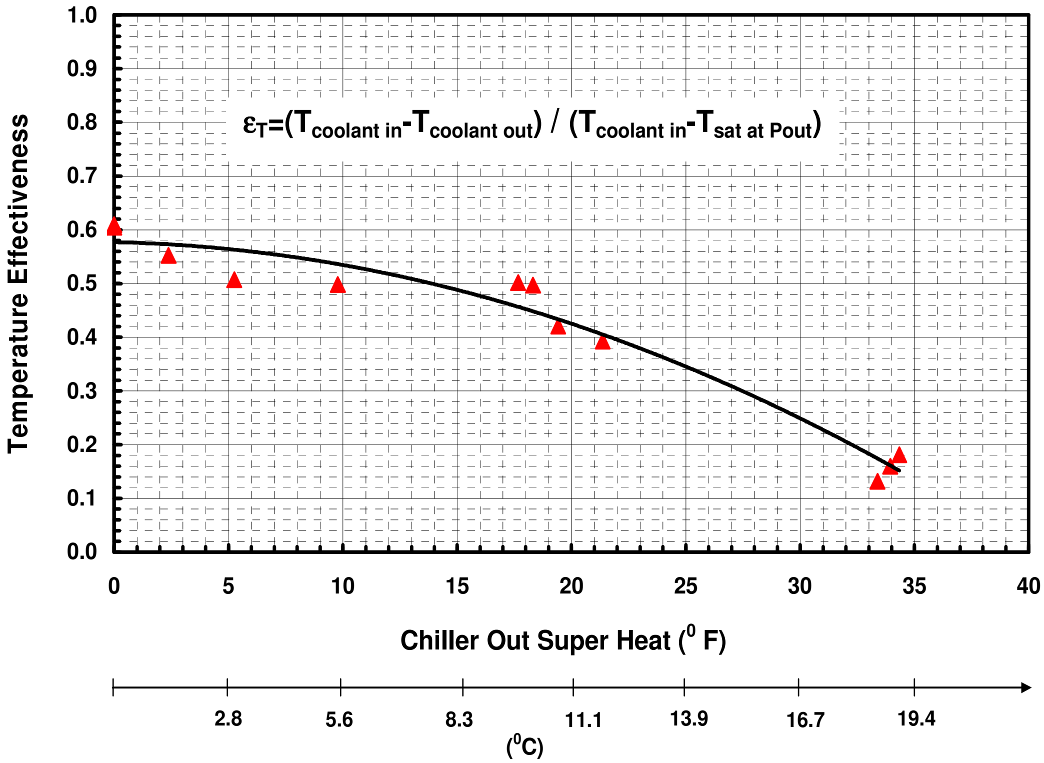

Figure 9 Effect of Refrigerant Chiller Out Super Heat on Chiller Effectiveness

6.5 Cooler

The cooler is a secondary fluid coolant to air heat exchanger. Physically it replaces the traditional evaporator inside the HVAC module.

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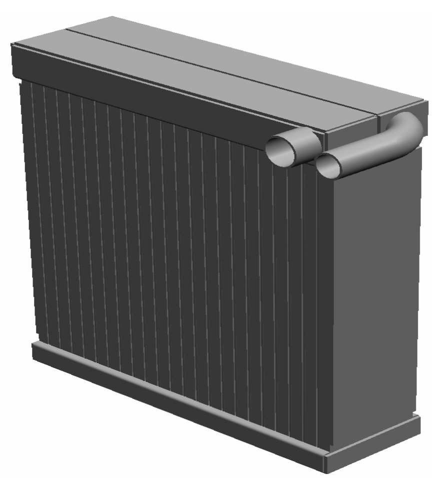
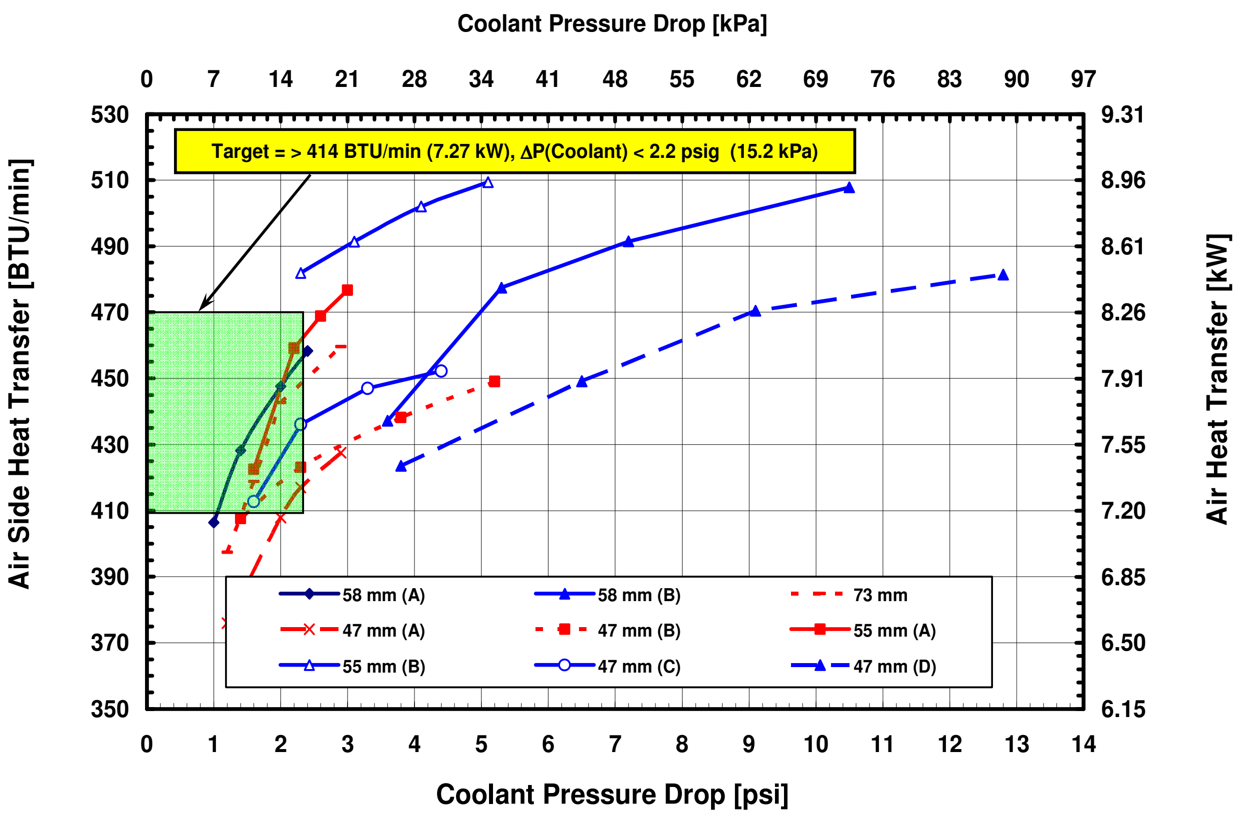


Figure 10 Typical Cooler

The secondary fluid (coolant) is chilled in the chiller and is pumped to the cooler where air is cooled to the desired temperature. Figure 11 illustrates the thermal performance of several coolers. Again, this heat exchanger has to be compact, lightweight and erosion / corrosion resistant. Such heat exchanger is shown in figure 9. It is aluminum brazed flat tube and center construction heat exchanger.



Air In: 100°F x 40% RH (38°C); Airflow Rate: 19.1 lbm/min (520 kg/hr)
Coolant In T: 30°F (-1.1°C); Coolant Flow Rate: 3, 4, 5, 6 gpm (11.4, 15.1, 18.9, 22.7 l/min)

Figure 11 Cooler Heat Transfer Performance

6.6 Pump

The coolant pump is a centrifugal type with plastic housing and impeller. It is a variable speed and magnetically coupled to a brush-less motor. The pump picture and its performance are shown in figure 12.

6.7 Secondary Fluid Selection (Coolant)

Proper selection of the secondary fluid (coolant) is key to the optimum performance of a secondary loop system. It is necessary for the secondary fluid to be non-flammable since it is circulated inside the vehicle passenger compartment. There are a variety of commercially developed, single phase heat transfer fluids which

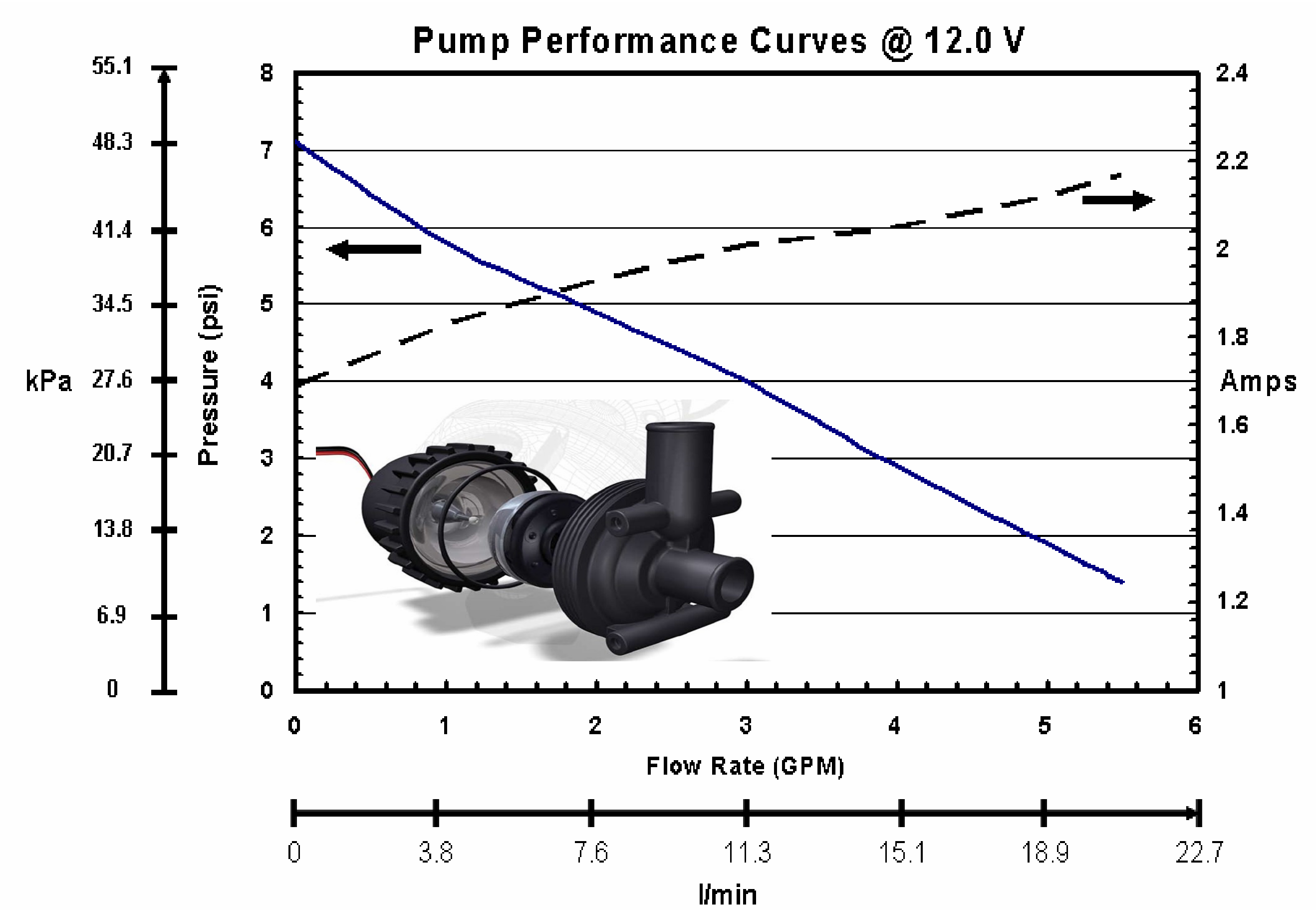


Figure 12 Coolant Pump

could be used as coolants. Glycols water mixtures, non-aqueous fluids such as synthetic organic fluids, Dowtherms, and Silicon oils are among the candidates. Because of their heat transfer degradation due their high viscosity at low temperatures, propylene glycol water solutions are mostly used in medium temperature applications above 0 ^{0}F (-18 ^{0}C). Ethylene glycol water solutions have better transport properties than their counterpart propylene solutions. Like any glycol solutions, increasing ethylene glycol concentration in water results in a performance penalty and an increase in pumping power. Therefore, the selection of the correct concentration for the secondary loop system applications is important. The lack of thermo-physical, safety, and automotive validation data coupled with the cost make the non-aqueous fluids not attractive at the moment.

The range of operating temperatures of a cooler in an automotive secondary loop air conditioning system is between 32 ^{0}F (0 ^{0}C) and 60 ^{0}F (15.5 ^{0}C), hence the evaluation of secondary fluid thermo-physical properties have to be taken into consideration in this temperature range. Based on heat transfer properties, familiarity, wide automotive validation and applications, and the presence of inhibitors to prevent corrosion, engine coolant which is ethylene glycol based was selected for this project. Figure 13 show the burst and freeze lines of the various glycol water concentrations.

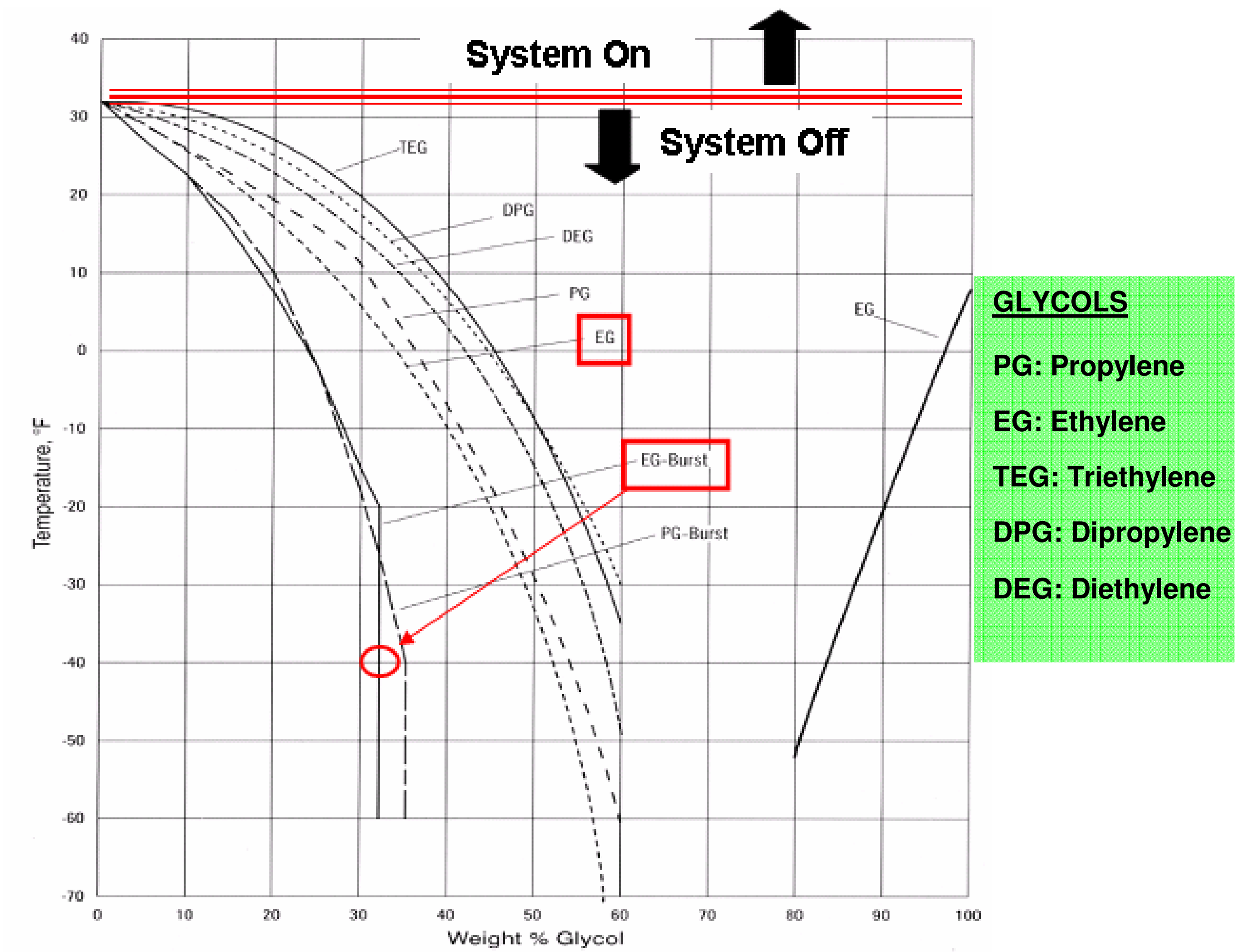


Figure 13 Secondary Fluids (Coolants)

A 32% ethylene glycol water solution exhibits adequate transport and thermo-physical properties and provides sufficient heat transfer while protecting the coolant circuit components from freeze damages. The EG "burst" line depicted in the figure represents the "zero expansion" at low temperatures when the solution is not pumped and the EG line stands for the glycol percent concentration if the solution is pumped through the coolant circuit. Generally automotive air conditioning systems are controlled not to operate when the ambient temperature is below 40 to 45 ^{0} F (4 to 7 ^{0}C), therefore the 32% ethylene glycol concentration is ample enough to protect the coolant components against very low ambient temperatures (~ -50 ^{0}F) when the air conditioning system is off.

VII. PERFORMANCE COMPARISON

The function of an A/C system is to quickly and efficiently maintain passenger's thermal comfort irrespective of external ambient conditions. The most severe conditions a vehicle is subjected to are the soak and cool down and city traffic at high ambients. In this section, the performance of R-152a secondary loop system is compared to a conventional R-134a direct system using the same vehicle and tunnel under identical testing conditions.

7.1 Test Description and Methodology

The vehicle was first tested with its original equipment for baseline purposes. The same vehicle was outfitted with a secondary loop system with same or similar in size as the current R-134a components such as condenser.

Table 1. Test Matrix

Ambient	A/C setting	Driving Conditions
115 ^{0}F x 15% R.H. (46 ^{0}C)	Vent Mode (O.S.A / Rec. Air, Hi Bl., F.C.) *	City Traffic Schedule
104 ^{0}F x 40% R.H. (40 ^{0}C)	Vent Mode (Rec. Air, Hi Bl., F.C.)	Soak & Cool Down; R.L.** Stable Points: 30, 50, 70 (48, 80, & 113 kph) & Idle
95 ^{0}F x 40% R.H. (37 ^{0}C)	Vent Mode (O.S.A., Hi Bl., F.C.)	R.L. Stable Points: 30, 50, 70 (48, 80, & 113 kph) & Idle
95 ^{0}F x 40% R.H. (37 ^{0}C)	Vent Mode (Rec. Air, Hi Bl., F.C.)	R.L. Stable Points: 30, 50, 70 (48, 80, & 113 kph) & Idle
80 ^{0}F x 60% R.H. (27 ^{0}C)	Vent Mode (O.S.A., Hi Bl., F.C.)	R.L. Stable Points: 30, 50, 70 (48, 80, & 113 kph) & Idle
59 ^{0}F x 70% R.H. (15 ^{0}C)	Vent Mode (O.S.A., Hi Bl., F.C.)	R.L. Stable Points: 30, 50, 70 (48, 80, & 113 kph) & Idle

*** O.S.A. = Outside Air; Rec. Air = Re-circulated (Cabin Air); Hi Bl. = High Blower; F.C. = Full Cold**
**** R.L. = Road Load**

Besides equipping the vehicle with standard instrumentation for climate control tunnel testing, the vehicle was also instrumented with several temperature sensors and pressure transducers to monitor all the relevant parameters and to extract maximum data information from both systems.

The performance comparison of the two systems were made under identical test conditions. The tests were performed in an identical environment using the same vehicular thermal tunnel. Table 1 lists the test conditions under which both systems were evaluated. The test matrix was designed to provide enough performance data for comparison purposes. It is based on testing procedures used for validation and benchmarking of production and prototype air conditioning systems. The first set of tests shown on the table were dedicated to high loads (high ambient) while the rest of the test matrix was devoted to capacity control for the purpose of reducing the energy usage by the secondary loop system.

7.2 Performance under High Ambient Conditions

In general, passenger comfort is related to the air conditioning discharge and cabin temperatures coupled with airflow and its circulation inside the passenger compartment. Figure 14 shows the dynamic changes of the discharge and cabin

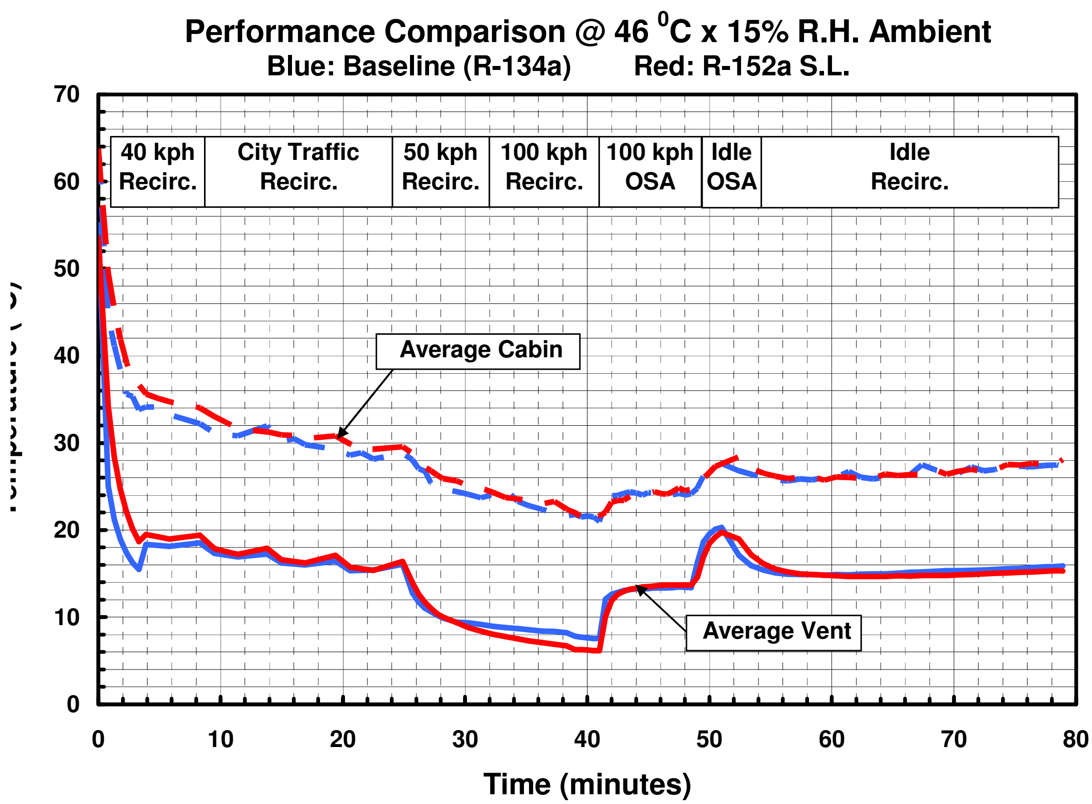


Figure 14 Performance Comparison at High Loads

temperatures under ambient conditions of 115 ^{0}F (46 ^{0}C) x 15% relative humidity. These conditions are representative of the Phoenix area summer climate. The driving cycle consisted of a soak, followed by a city traffic schedule in re-circulated mode and finally an idle in outside air mode.

As seen in the graph, at various driving conditions the behavior of the vent discharge and cabin temperature variations between the secondary loop system and the baseline production system are almost identical through the driving cycle regardless of supplied air, being in outside air (fresh air) or re-circulated air (cabin air). Figure 15 presents the cooling performance comparison of the two systems at 104 ^{0}F (40 ^{0}C) x 40% relative humidity under soak and cool conditions, and stable driving conditions. Again, the discharge and cabin temperatures are almost identical through the driving cycle.

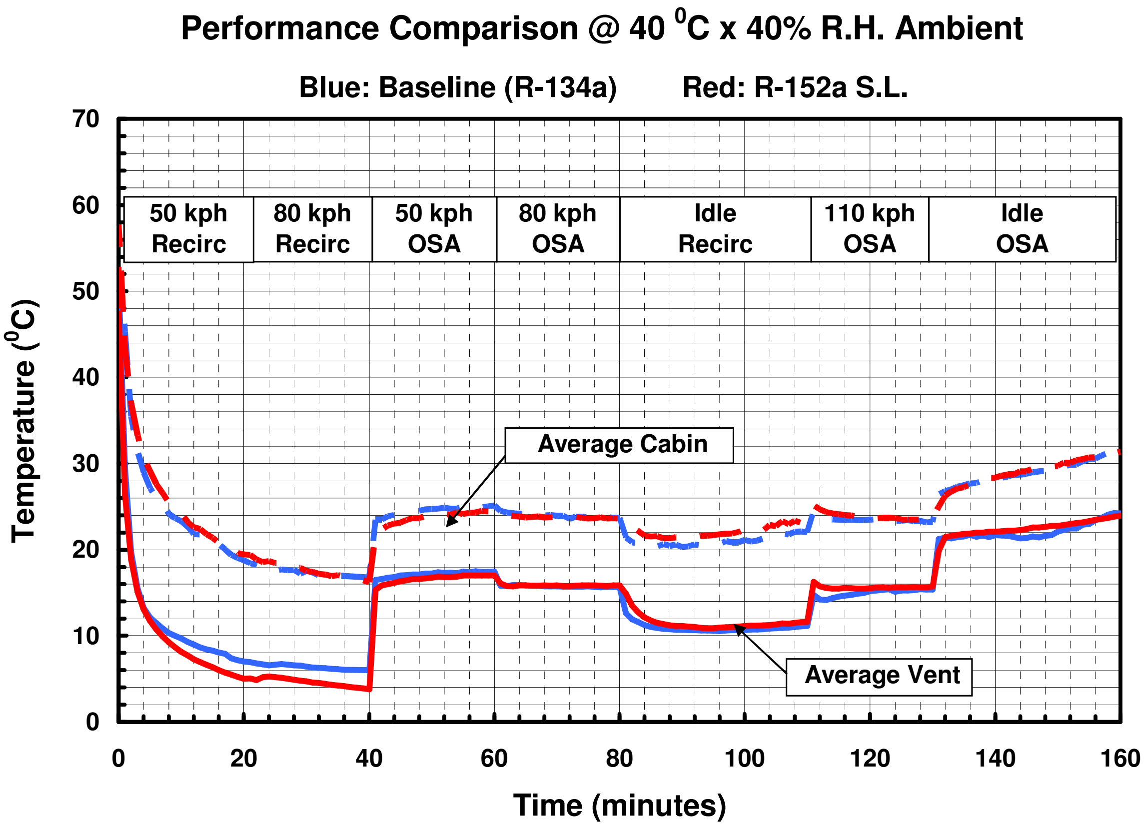


Figure 15 Performance Comparison @ High Loads

The secondary loop system response time during soak and cool conditions sometimes called “thermal lag” was originally a concern because of the thermal capacitance associated with the coolant. However, as shown in figures 14 and 15, both systems were soaked to the same temperature, yet the response time delay (thermal lag) of the secondary loop system is negligible. The thermal lag concerns are emanated from thoughts that the coolant temperature would match the soak temperature (cabin temperature) when the vehicle is soaking in the sun. However, data has shown that the coolant temperature stays at ambient temperature since the whole system including the coolant reservoir is under-hood and not subjected to the effect of the sun load. Over the years, the results of numerous subjective evaluations of the secondary loop carried out under high ambient conditions concluded that the thermal lag is not an issue. The thermal lag is certainly measurable however not perceptible. Some of the recent subjective evaluation is described in the following section.

7.3 Road Performance under High Loads

One of the four demonstration vehicles shown at the 2007 Alternate Refrigerant Symposium was equipped with an R-152a secondary loop system.

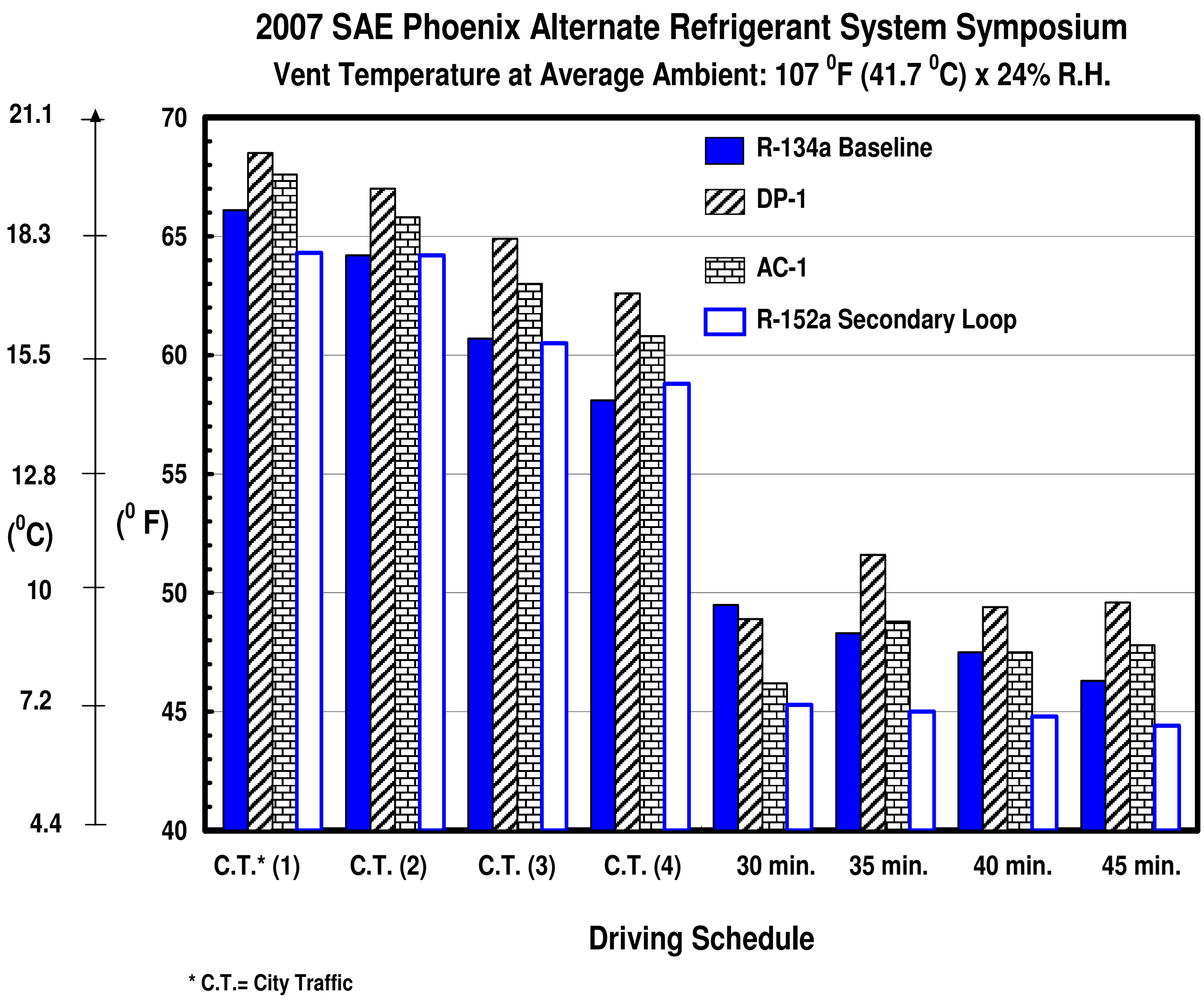


Figure 16 Road Performance at High Loads

The forum [6] was held in Scottsdale, Arizona and was represented by almost 270 attendees. The participants came mainly from the mobile air conditioning industry. Besides discussing the progress of alternate refrigerant technologies, the forum covered the comfort evaluation of the demonstration vehicles. The evaluation consisted of a subjective comfort rating coupled with air vent and cabin temperature readings at each phase of the driving cycle.

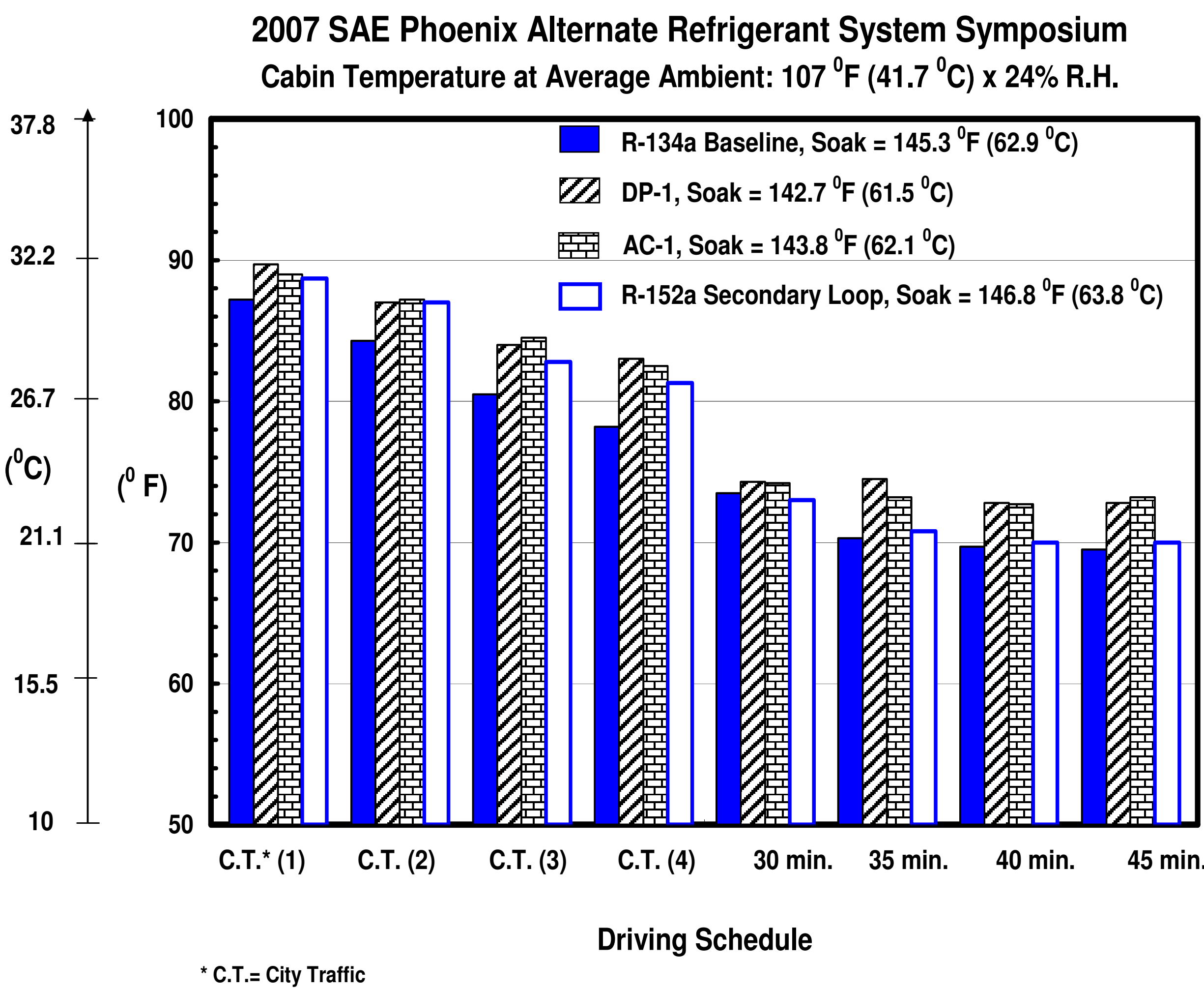


Figure 17 Road Performance at High Loads

The demonstration vehicles were identical (2007 Opel Astra) including exterior and interior colors. The assessors were not informed on what type of refrigerant or system is in each vehicle (blind evaluation). The rides were random and blind for the purpose of eliminating any bias for or against a typical refrigerant or system. Figures 16 and 17 display the objective evaluation, which is represented by the measurements of the vent and cabin temperatures of the R-134a baseline, DP-1 refrigerant system, AC-1 refrigerant system, and R-152a secondary loop system. This objective ratings show that the baseline and the R-152a secondary loop system are similar in performance and are better performers than DP-1 and AC-1 systems.

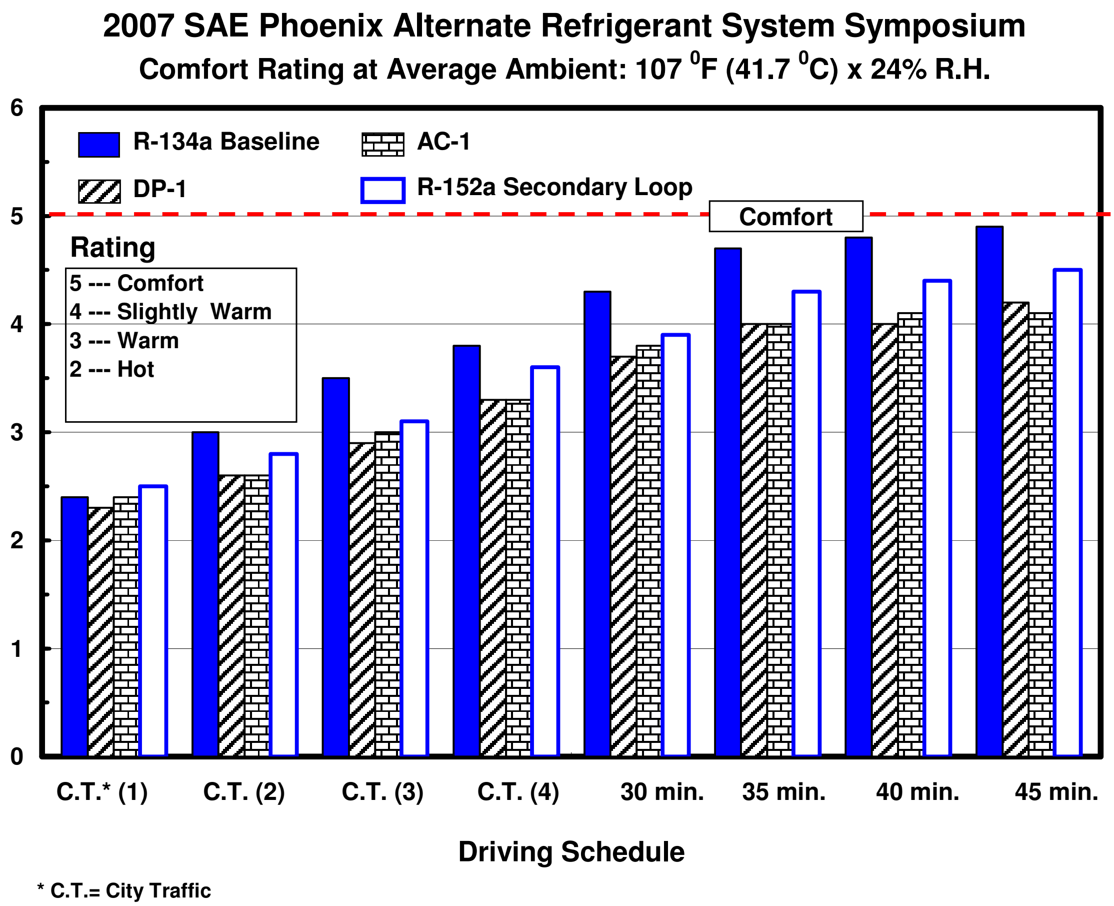


Figure 18 Subjective Comfort Evaluation at High Loads

The results of the subjective comfort rating are shown in figure 18. They follow similar patterns as the objective evaluation. Over all the performance of the R-152a secondary loop system matches the baseline system.

7.4 Performance under Medium to Mild Ambient Conditions

Mobile air conditioning systems are generally designed and sized for maximum loads which in most cases are soak and cool downs under extreme environmental conditions. Under medium ambient conditions such as 95 ^{0}F (35 ^{0}C) x 40% relative humidity and mild ambient conditions such as 80 ^{0}F (27 ^{0}C) x 60% relative humidity, overcooling occurs in nearly all vehicles resulting in a reheating process by tempering with the temperature door, sometimes called blend door to maintain comfort. This occurrence is sometimes called “series reheat” in the industry. In order to reduce the series reheat, today’s systems require sophisticated and expensive controls to operate

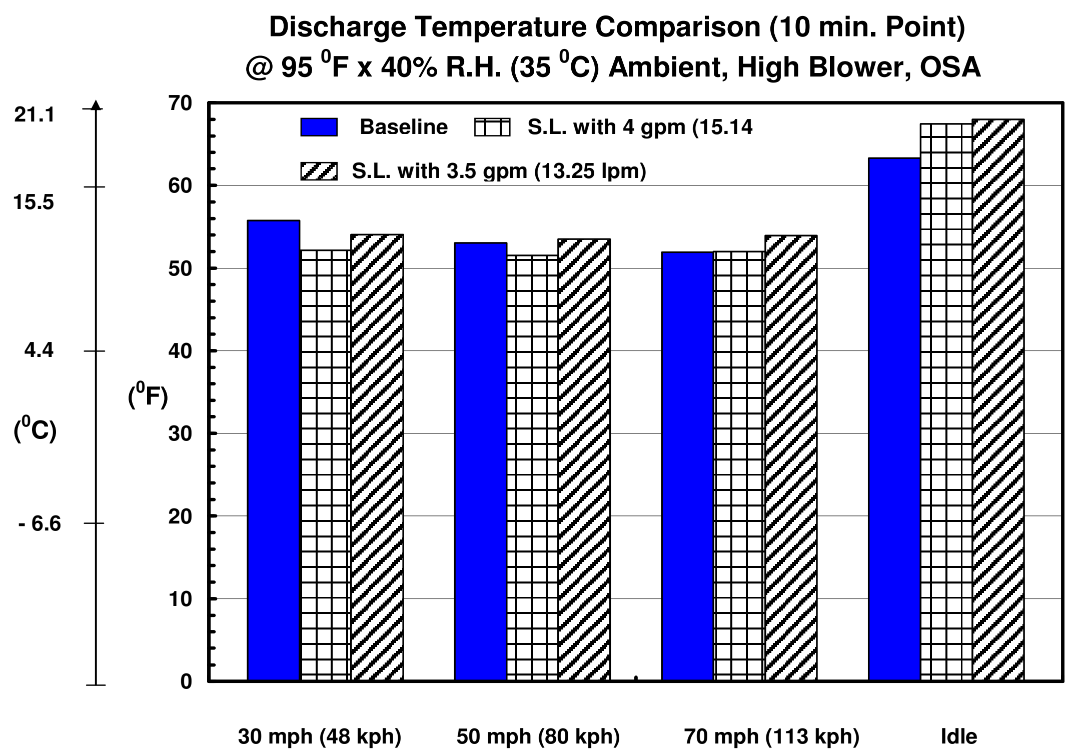


Figure 19a Overcooling Reduction

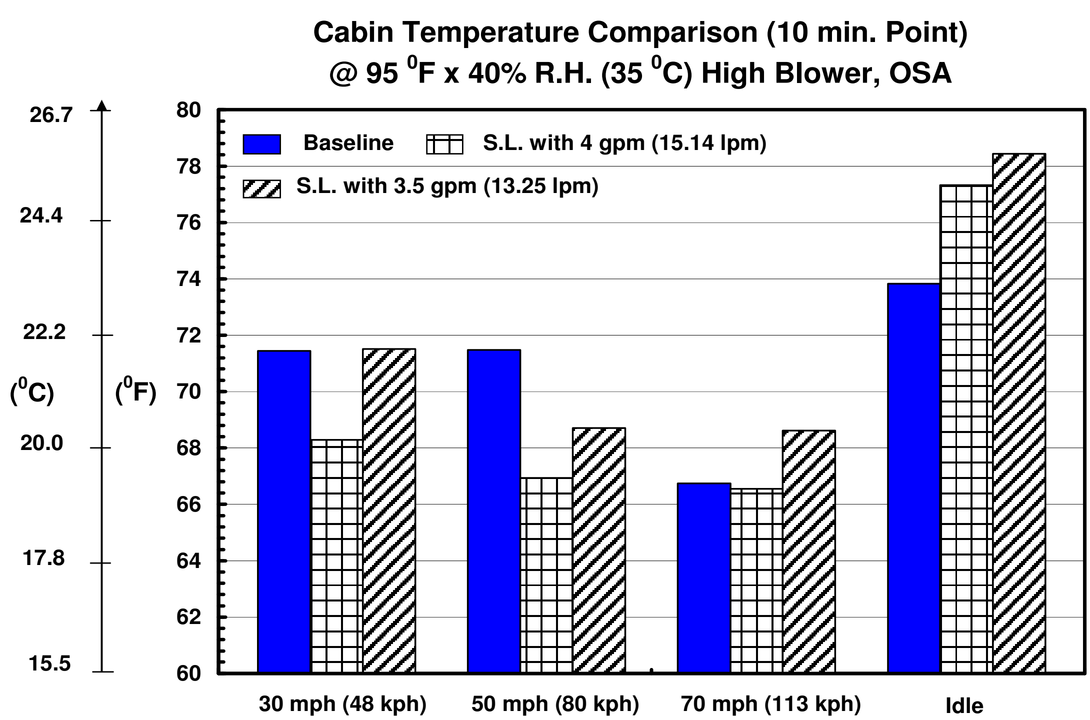


Figure 19b Overcooling Reduction

the evaporator at higher pressures. In a conventional system, operating the evaporator at higher pressures elevates the evaporator air outlet temperature thus minimizing the reheating process and in turn reducing the compressor shaft power.

The secondary loop system offers this benefit of overcooling prevention by simply varying the coolant flow using the speed of the pump, which in turn, reduces the compressor shaft power. This technique of "series reheat" reduction is not complicated and is less costly.

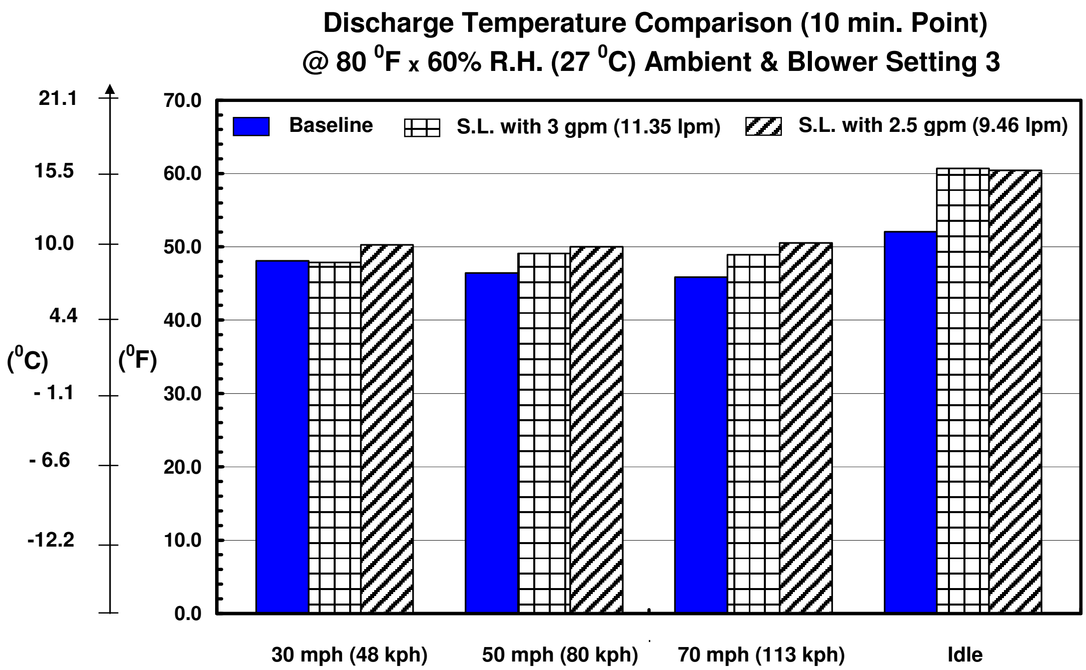


Figure 20a Overcooling Reduction

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Figures 19a and 19b depict a typical wind tunnel test data that represents the cooling performance comparison of the baseline and the secondary loop system at 95 ^{0}F (35 ^{0}C) x 40% relative humidity. The tests were conducted at vehicle speeds of 30, 50, 70 mph (48, 80, 113 kph) and idle. In the data shown in figures 19a and 19b, the blower was set to high position while the air supply was dialed in outside air mode (fresh air). The airflow rate in outside air mode at high blower position is around 260 ft^3/min. (7.36 m^3/min.) for both systems. In re-circulated air mode at blower setting 3 which represents high medium blower, on the average the airflow rate for the secondary loop system is higher than the baseline. The average amount of air flow rate in re-circulating mode for the secondary loop system at blower setting 3 is 216 ft^3/min. (6.12 m^3/min.) while the baseline air flow rate is 205 ft^3/min. (5.80 m^3/min.). Table 2 shows the airflow rates at various HVAC blower settings. The variation among the airflow rates between the baseline and the secondary is mainly due to the depth difference between the baseline evaporator and the secondary loop cooler.

Table 2	Outside Air (OSA)		Re-circulated Air (Recirc.)	
	R-134a	S.L.	R-134a	S.L.
	ft^3/min (m^3/min)	ft^3/min (m^3/min)	ft^3/min (m^3/min)	ft^3/min (m^3/min)
Blower Setting 2	135 (3.82)	140 (3.96)	147 (4.16)	155 (4.39)
Blower Setting 3	188 (5.32)	198 (5.61)	205 (5.80)	216 (6.12)
High Blower	263 (7.45)	260 (7.36)	286 (8.10)	281 (7.96)

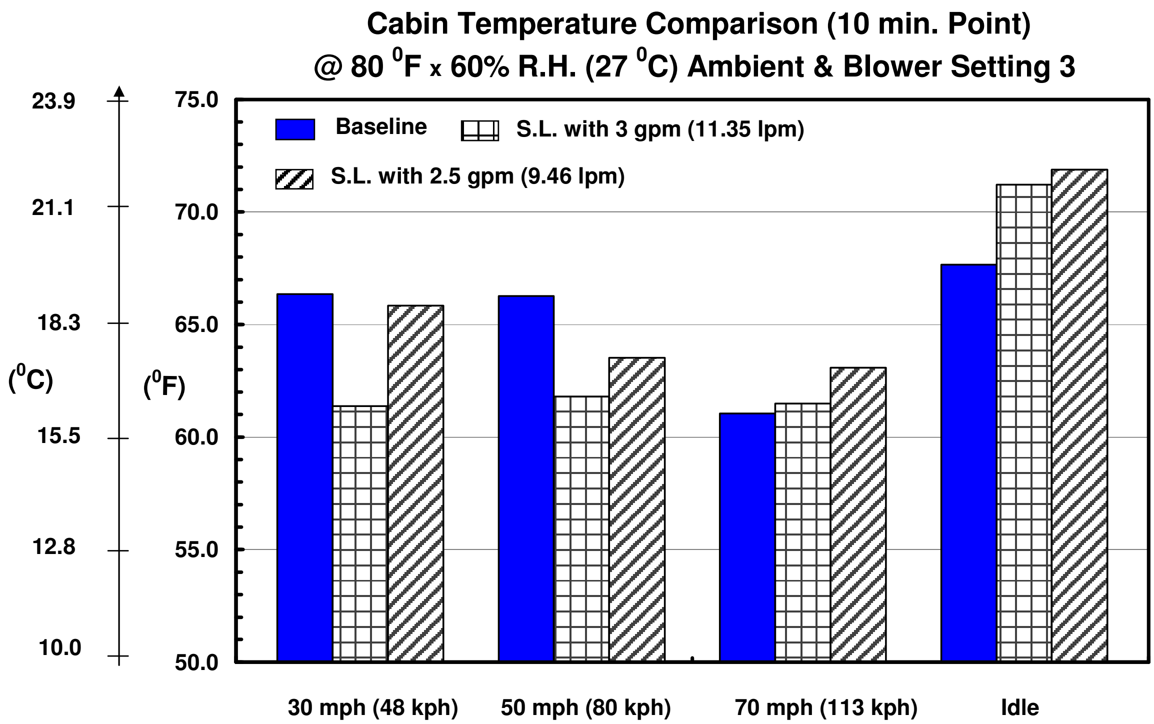


Figure 20b Overcooling Reduction

As shown in graph 19a, when the coolant pump is run at full capacity (4 gpm / 15.14 lpm), on the average the secondary loop vent discharge temperatures are lower than the baseline temperatures. Dialing down the coolant flow rate to 3.5 gpm, (13.25 lpm) raised the vent temperatures up to par with baseline or higher, thus avoiding overcooling. After long driving periods, in current mobile air conditioning systems without series reheat reduction option, overcooling may take place and most vehicle operators would dial up the temperature door (blend door) for reheating purpose. However, this maneuver consists mainly of redirecting a certain percentage of the cold air toward the heater core for warmth and later on blends it with the outgoing vent air. This process does not reduce or influence the load on the system hence it does not diminish the compressor work.

Instead of relying on temperature door (blend door) movements for reheat, the secondary loop system uses the coolant flow variation as the controlling parameter. This reheating process is accomplished by adjusting the coolant pump speed (decreasing coolant flow) which in turn lowers the load on the system and hence reducing the compressor shaft work. The compressor work reduction is achieved by cycling off a fixed displacement compressor or de-stroking a variable displacement compressor. Figure 19a illustrates the reheating process reduction in outside air mode by turning down the coolant from 4 gpm (15.14 lpm) to 3.5 gpm (13.25 lpm). Conventionally in a baseline, system with the reheating option or in a secondary loop system with modulating coolant flow for reheating purposes the passenger comfort is always respected and not compromised.

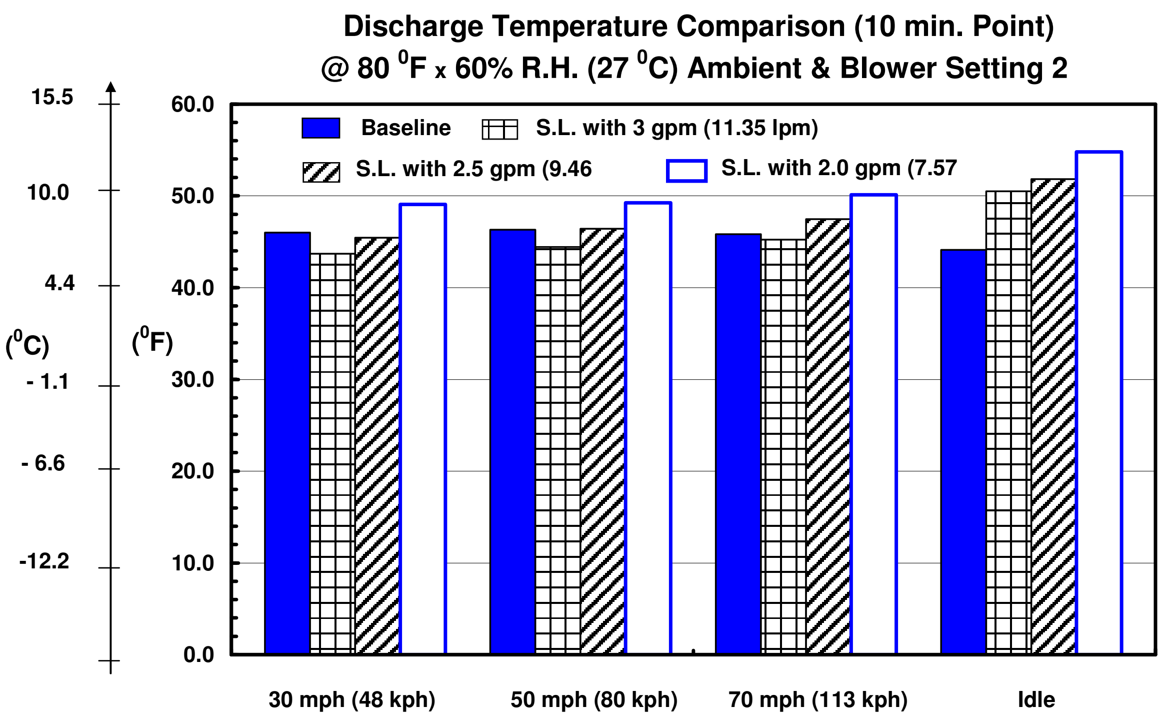


Figure 21a Overcooling Reduction

In today's air conditioning systems, under mild ambient conditions such as 80 ^{0}F (27 ^{0}C) x 60% relative humidity and low ambient conditions, overcooling very frequently occurs and in most cases to maintain comfort, reheating using temperature door (blend door) levels and/or airflow reduction is needed.

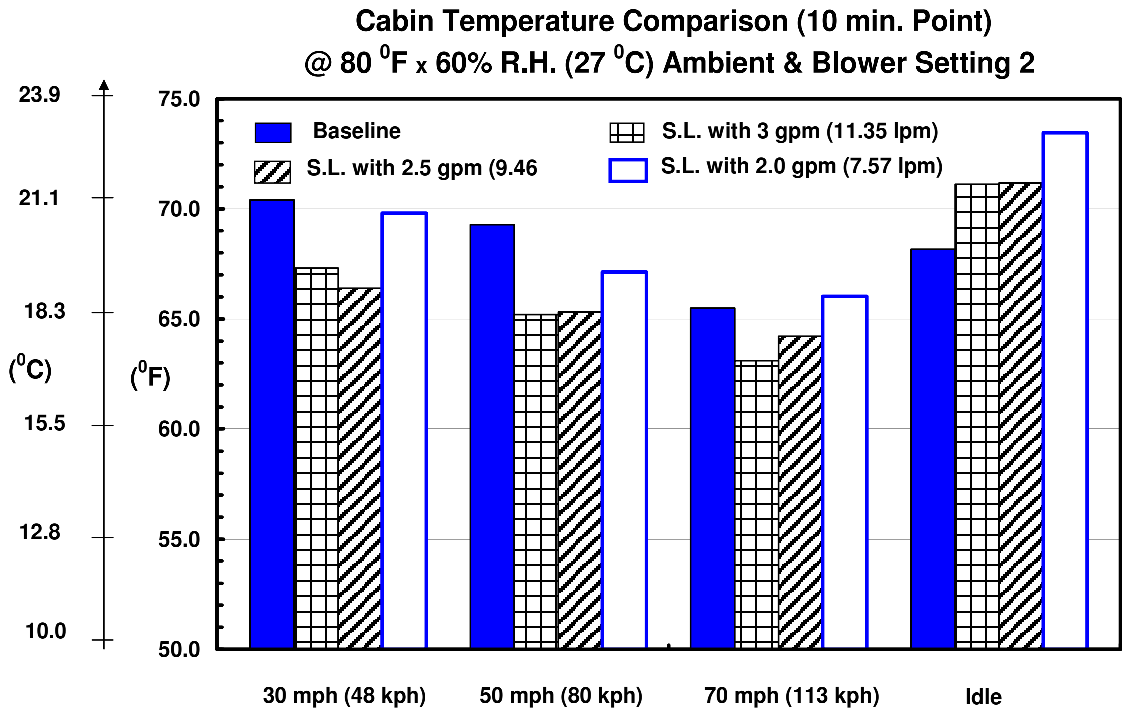


Figure 21b Overcooling Reduction

Figures 20a through 21b show the cooling performance comparison of the baseline and the secondary loop at 80 ^{0}F (27 ^{0}C) x 60% relative humidity with air supply set in outside air mode (fresh air). At this mild ambient and below, most vehicle operators prefer fresh air versus cabin air unless somehow the outside fresh air is contaminated. As previously stated, the tests were conducted at vehicle speeds of 30, 50, 70 mph (48, 80, 113 kph) and idle.

Figures 20a and 20b describe the discharge and cabin temperatures under moderate loads portrayed by high medium blower (setting 3) and outside air conditions. Instead of operating the pump at maximum coolant flow, which is in this case over 4 gpm (15.14 lpm), the pump is operated at 3 or 2.5 gpm (11.35 or 9.46 lpm). The graphs show that overcooling reduction can be achieved by operating the pump at 2.5 gpm (9.46 lpm) or below without compromising the comfort. Under lower loads where the blower is set at medium (setting 2) or lower, the coolant is cut down further to 2 gpm (7.57 lpm) or lower, therefore reducing the load on the primary system and consequently the compressor shaft work is reduced. The load on the primary system, which is mainly the compressor power, is brought down by reducing the overcooling depending on the desired comfort. Figures 21a and 21b present an array of load reduction on the system by means of low coolant flow rates when the A/C demand is not high (low load).

VIII. ENERGY COMPARISON

The coefficient of performance (COP) comparison was carried under 104 ^{0}F (40 ^{0}C) x 40%, 95 ^{0}F (37 ^{0}C) x 40%, 80 ^{0}F (27 ^{0}C) x 60% relative humidity. The COP discussed in this section is the thermal COP, which is defined as the ratio of the refrigerant cooling over the compressor power.

$$COP = \frac{h_{out\,(Heat\ Exchanger)} - h_{in\,(Heat\ Exchanger)}}{h_{out\,(Compressor)} - h_{in\,(Compressor)}}$$

Where:

Heat Exchanger represents the evaporator and chiller for baseline and secondary loop respectively, and h is the refrigerant enthalpy Btu / lbm (kJ/kg) calculated at measured inlet and outlet pressure & temperature of each component.

8.1 Energy Comparison at Full Capacity

As depicted in figure 22, at the same capacity when compared to the baseline, the secondary loop system has on the average a lower COP. The outcome was

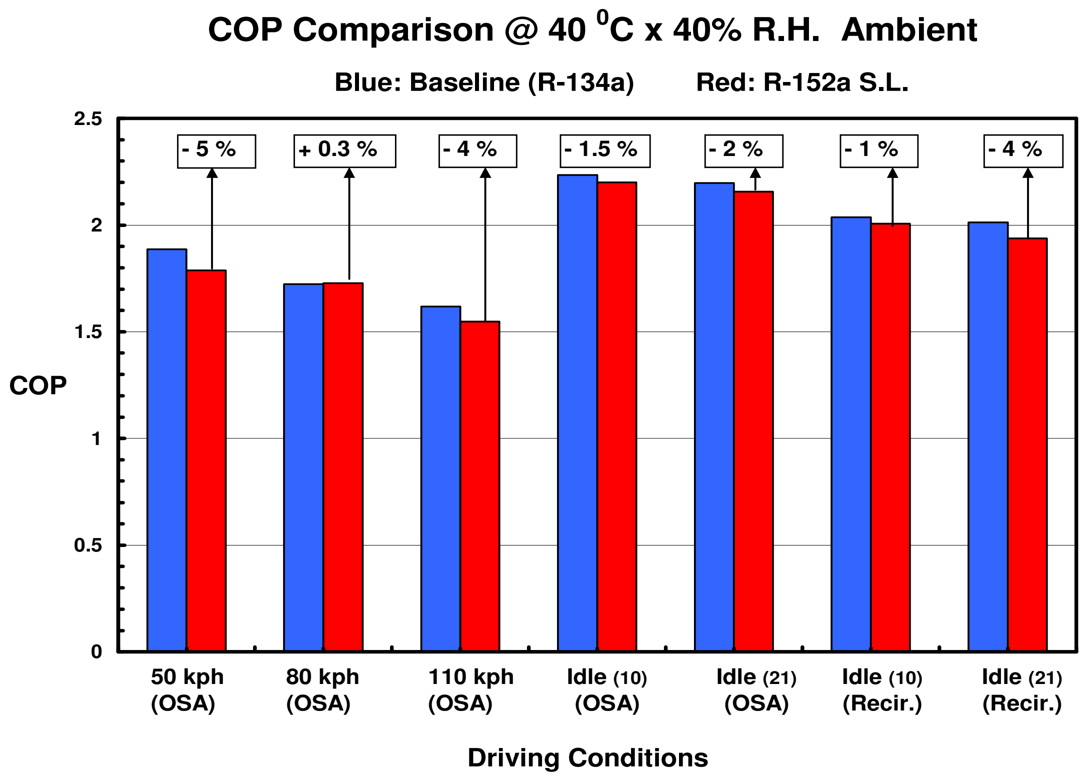


Figure 22 Coefficient of Performance

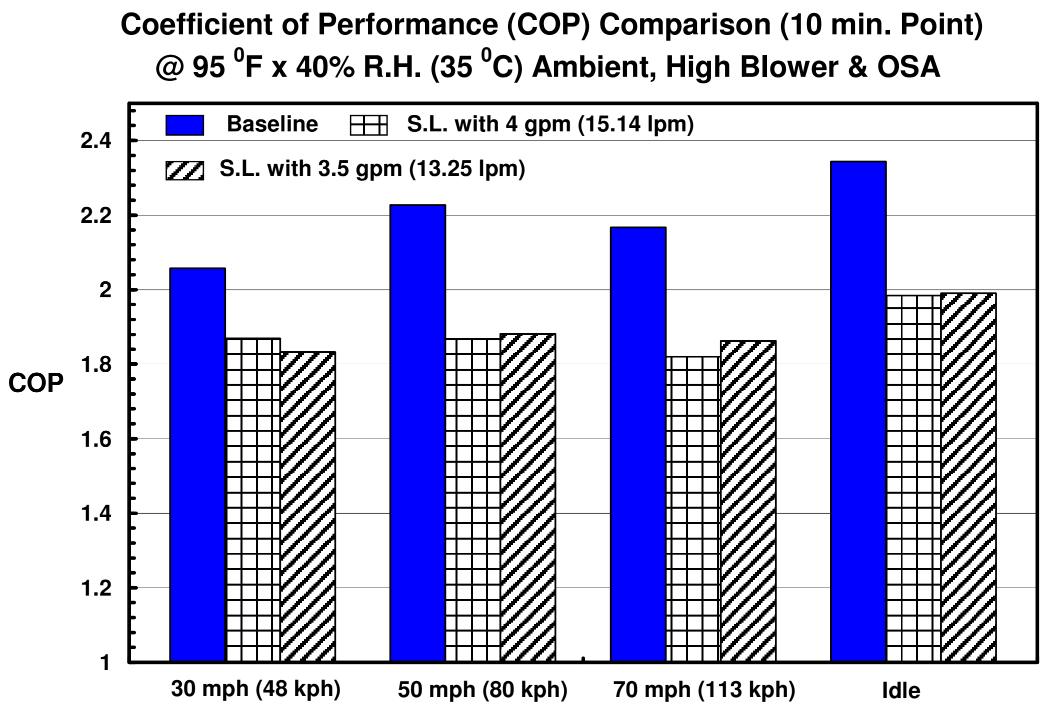


Figure 23 Coefficient of Performance (COP) Comparison

expected and was proven before by the author [4] in his study of secondary loop. The COP percentage difference between the baseline and the secondary loop is lower in this investigation than the previous study. This is attributed to R-152a being a better refrigerant than R-134a. As we all know the coefficient of performance (COP) is the ratio of cooling capacity versus the energy usage of a system. In more simple terms, it is what you get in cooling benefit or comfort versus (useful effect) over what you pay for (energy that must be purchased) to get the benefit. Technically the COP cannot be labeled as efficiency since it is higher than one, however it is a measure of how environmentally or economically efficient the cooling (comfort) is attained. The COP is not a measure of energy usage, it is more like how cost-effective or otherwise the energy is paid for through fuel economy or emission.

8.2 Energy Comparison at Controlled Capacity

Figures 23 through 24b compare the coefficient of performances (COPs) of the baseline system with several control capacity options of the secondary loop system at medium and low loads. The graphs clearly show that the COP gap between the baseline and secondary loop is widening as the load is decreased. This behavior was also observed by the SAE report on improved efficiency of mobile air conditioning (IMAC) [8]. The lower COPs for the secondary loop system are attributed to two factors, the lower compressor efficiency at reduced compressor strokes and compressor operation with R-152a instead of R-134a.

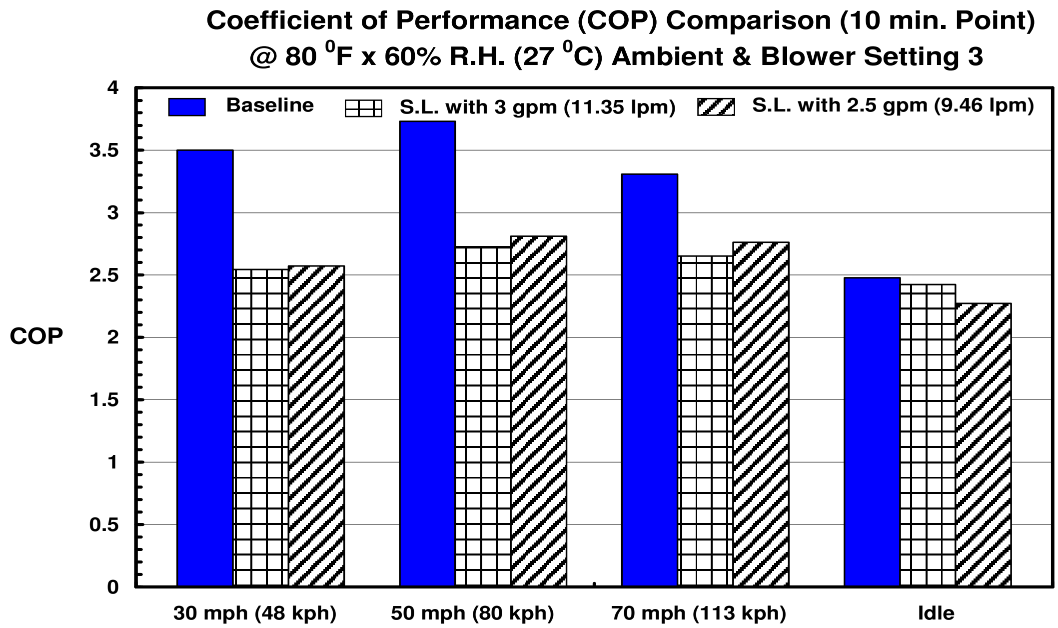


Figure 24a Coefficient of Performance (COP) Comparison

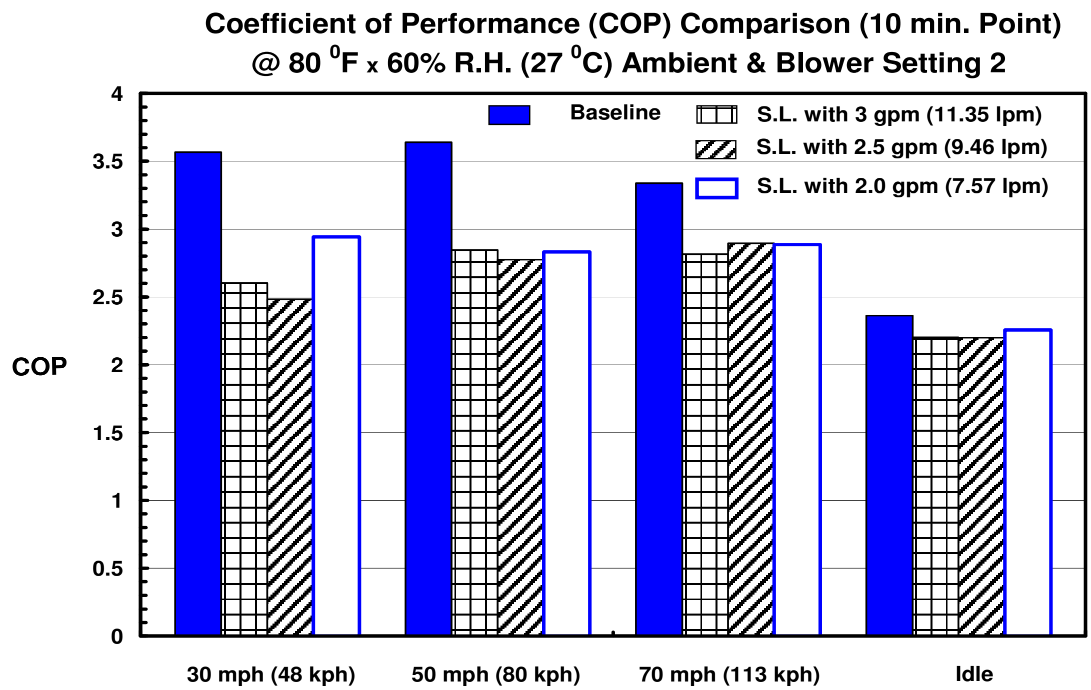


Figure 24b Coefficient of Performance (COP) Comparison

As in IMAC study, the energy usage or compressor power is a better metric than COP to evaluate fuel usage and emissions. The cooling capacity is directly proportional to the cooler (evaporator) air out temperature. The capacity reduction is accomplished by increasing the cooler (evaporator) air out temperature while keeping the comfort at the desired level. For the secondary loop system, the increase of cooler air out temperature is achieved by means of reducing the coolant flow rate.

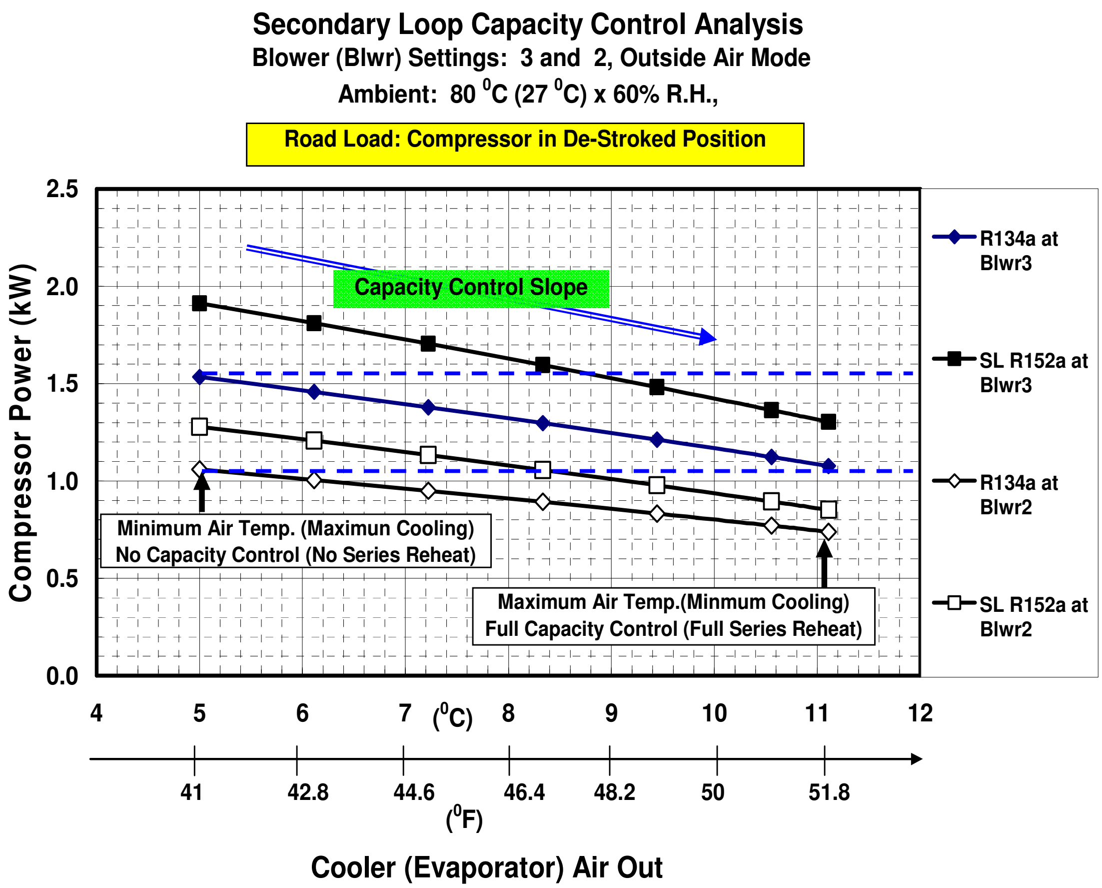


Figure 25a Compressor Power versus Capacity at Rod Load Conditions

Figures 25a and 25b describe the compressor power variation with the cooler (evaporator) air out temperature. The data points on the left and the right of the graphs illustrate the maximum cooling capacity (lowest cooler / evaporator air out) and the minimum cooling capacity (highest cooler / evaporator air out) respectively. It is evident from the graphs, at identical cooling capacities; the secondary loop system compressor uses more power than the baseline compressor. Most of the time, at medium to low loads, operating at maximum cooling capacity is not necessary. It initiates overcooling and as mentioned previously a reheating is needed to maintain comfort. Operating the secondary loop system below the blue dashed lines shown on the graphs will reduce overcooling. The capacity is controlled to match the desired comfort without relying on reheating. As shown in figure 25a at road load, if the cooler air out temperature is controlled to 46 ^{0}F (8 ^{0}C) or above the secondary loop system with controlled capacity will use less compressor shaft power than the baseline. At idle, the control temperature is 49 ^{0}F (9.5 ^{0}C). As seen on the plots, the secondary loop system compressor power can be maintained at equal or lower values of the baseline compressor power by means of controlling the cooling capacity.

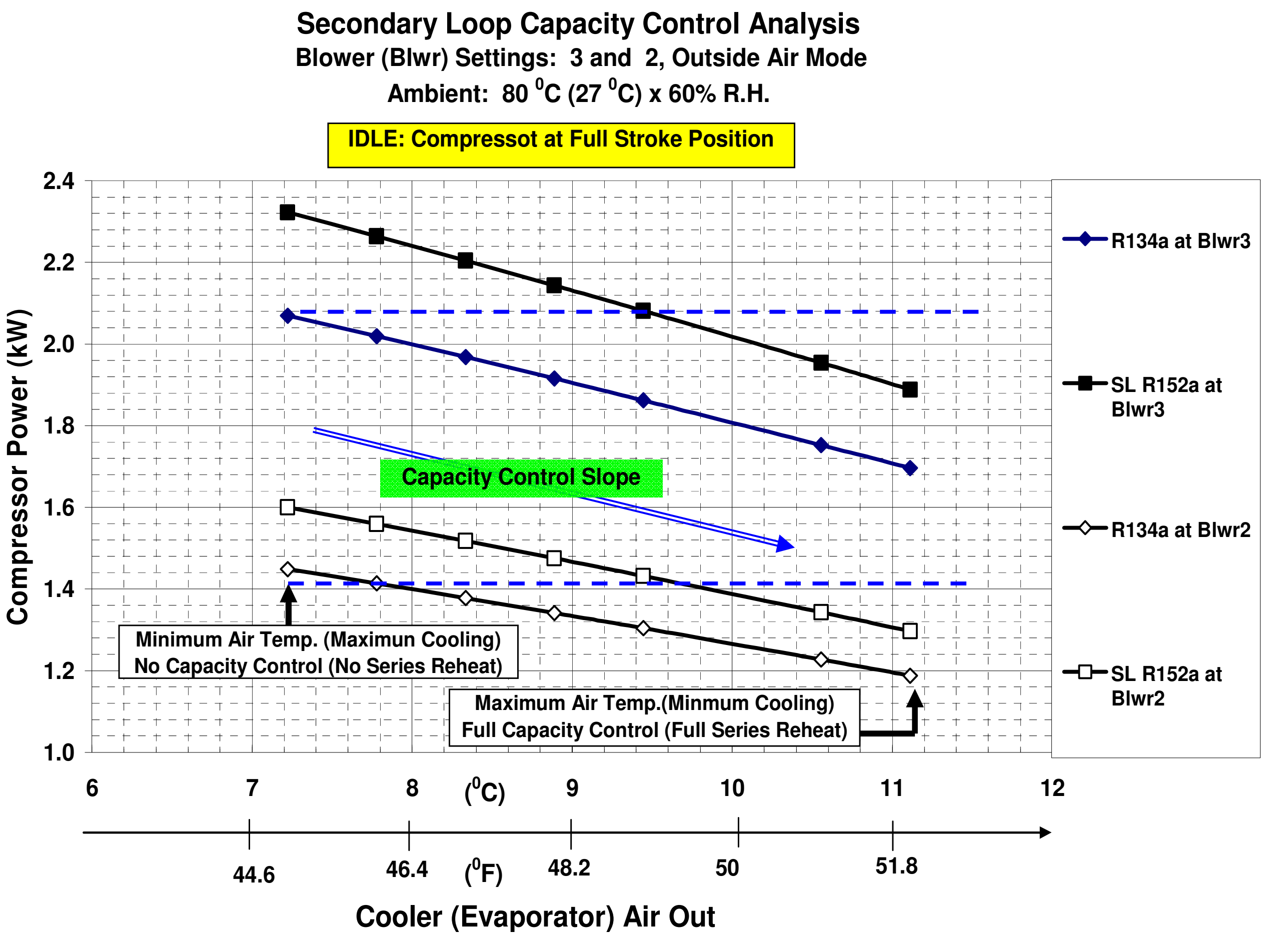


Figure 25b Compressor Power versus Capacity at Idle Conditions

As mentioned before, the capacity control is achieved through modulating the coolant flow rate. At medium to low loads, the secondary loop system capacity control is an effective way of reducing the compressor shaft power. The next section addresses the benefit in fuel consumption generated through the reduction in compressor shaft power.

8.3 Fuel Consumption Analysis

The objective of this section is to estimate the air conditioning fuel use of a baseline R-134a and a secondary loop R-152a for various driving schedules. National Renewal Energy Laboratory (NREL), because of their experience and expertise in this field has cooperated with Delphi to carry out the fuel usage estimation. John Rugh, a senior engineer at NREL, has championed the analysis.

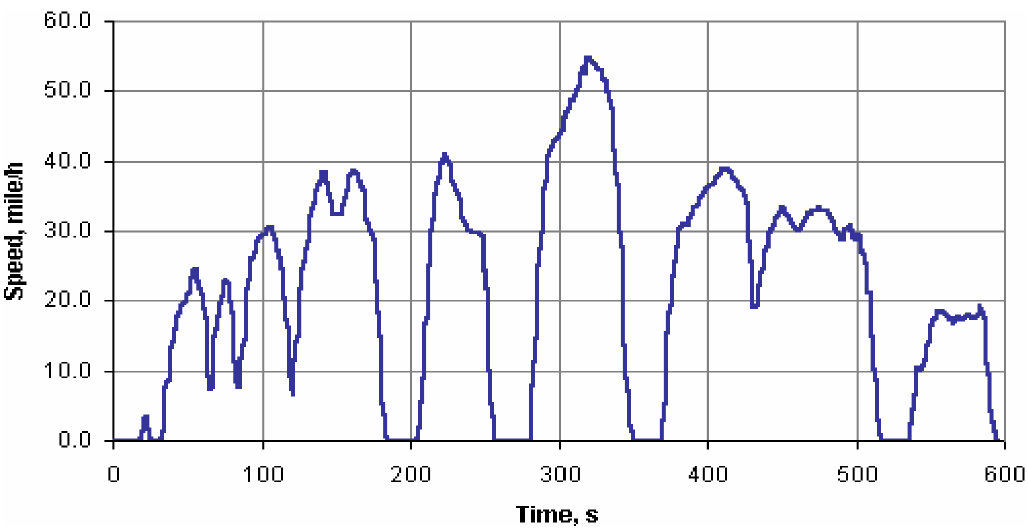


Figure 26a Speed Correction Schedule (SC03)

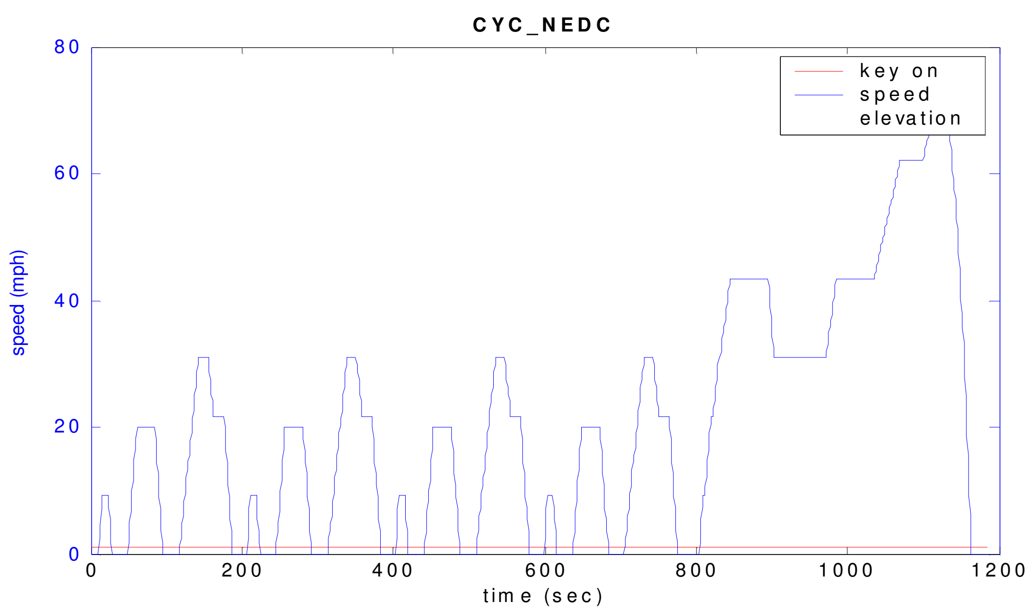


Figure 26b New European Driving Cycle (NEDC)

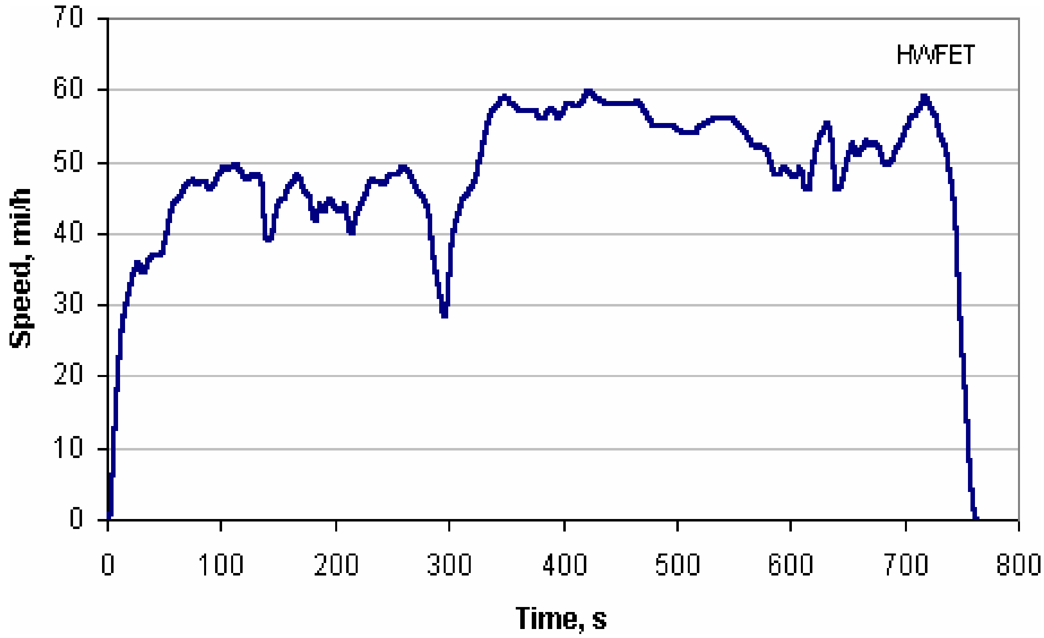


Figure 26c Highway Fuel Economy Test

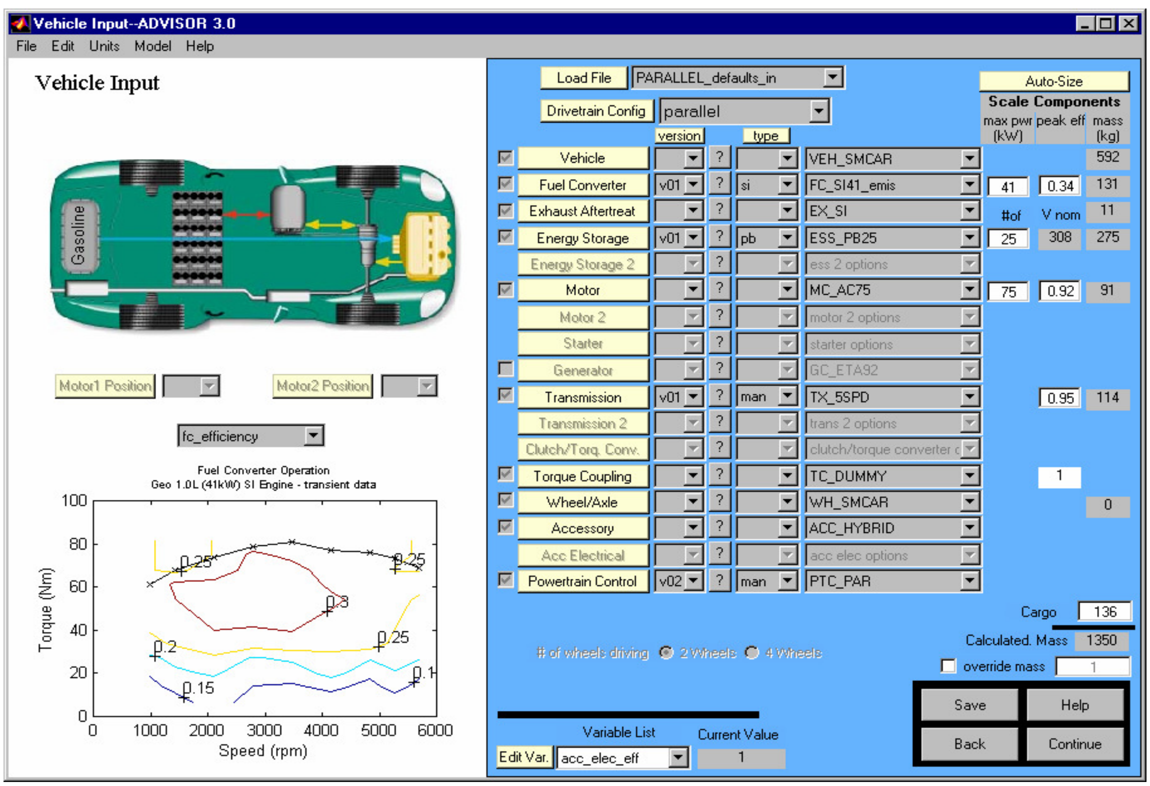
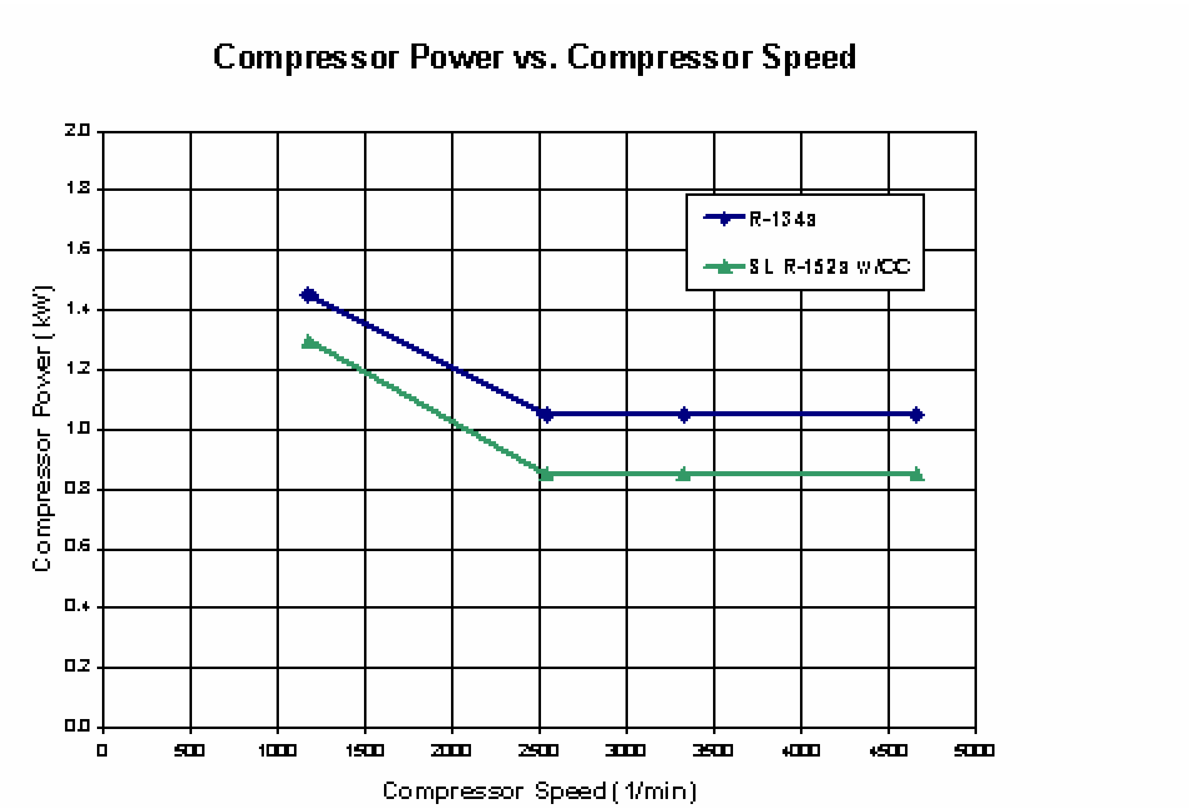


Figure 27 Sample of Inputs to the "Advisor" Software

The inputs to the model consisted of the compressor power versus speed generated from vehicle tunnel tests at ambient of 80 ^{0}F (27 ^{0}C) x 60% relative humidity, the vehicle characteristics such as mass and engine type, and the drive cycle data. The driving schedules simulated in this analysis are the New European Driving Cycle (NEDC), the Speed Correction Cycle (SC03), and Highway Fuel Economy Test (HWFET). These driving schedules depicted in figure 26a through 26c are run with the A/C off and A/C on. The vehicle model is built using "Advisor Software" with inputs such as coefficient of drag, frontal area, mechanical accessory load, and rolling resistance coefficient and engine characteristics. A sample of these inputs to the software is shown in figure 27. The vehicle and A/C data are shown in figure 28.



Coefficient of Drag: 0.32; Frontal Area: 2.11 m2 (22.7 ft2) Mass: 1263 kg (2784 lbm), R134a 1270 kg (2799 lbm), SL152a A/C off Mechanical Accessory Load = 500 W
Rolling Resistance Coefficient = 0.009
Engine – 63 kW 1.9 L Saturn Engine Scaled to 75 kW

Figure 28 Vehicle and A/C Data

The vehicle A/C data was taken under the ambient of 80 ^{0}F (27 ^{0}C) x 60% relative humidity, vent mode, full cold and the blower setting was at position 2 which corresponds to low medium blower (~ 6 volts). The coolant pump power of 15 Watts was taken into account in the analysis.

Figure 29 shows the total fuel consumption per 100 km of the vehicle with the air conditioning off, with R-134a system on, and with the R-152a secondary loop system with control capacity on. As expected the total fuel consumption with A/C off at each driving cycle is the lowest, however with A/C on, the consumption is less with the control capacity secondary loop system than the baseline R-134a system.

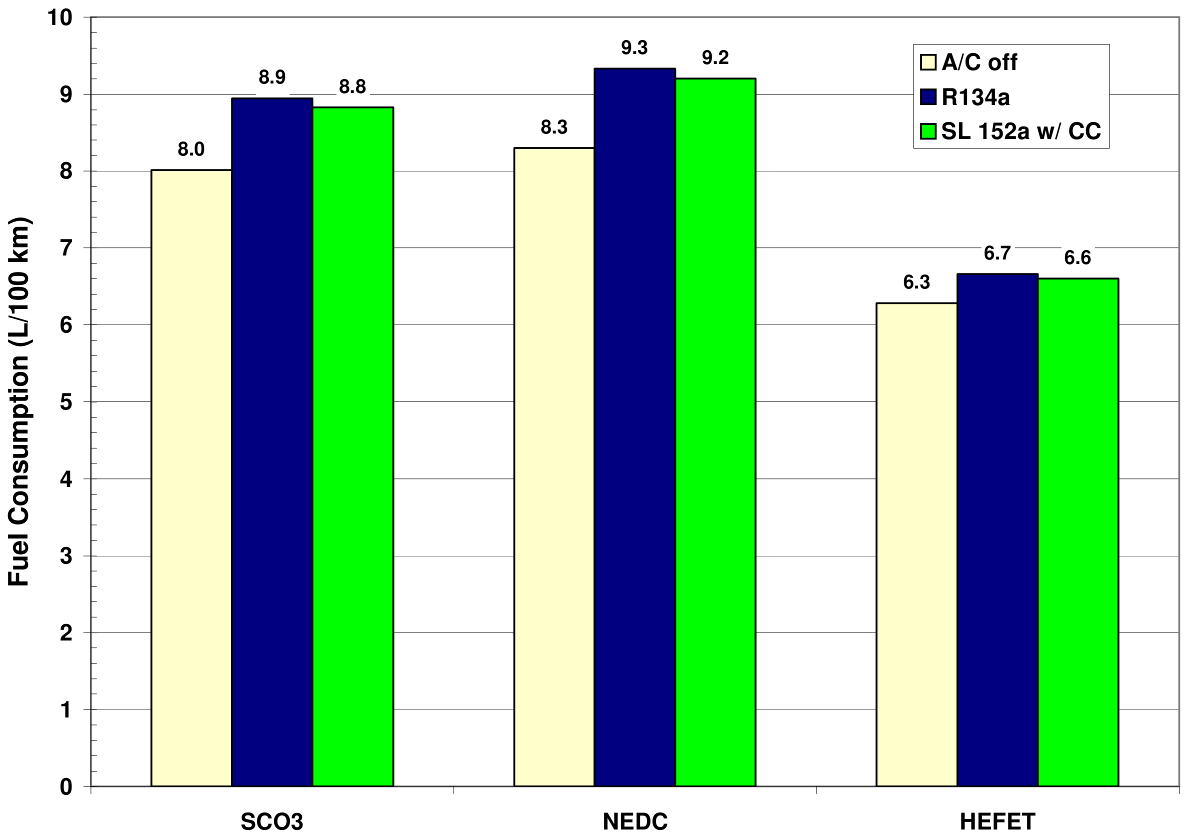


Figure 29 Overall Vehicle Fuel Consumption with A/C Off and A/C on

Figure 30 shows the air conditioning fuel usage per 100 km comparison between the baseline R-134a system and the R-152a secondary loop system with controlled capacity. Depending on the driving cycle, the advantage of the secondary loop with controlled capacity ranges from 13 to 16%. This fuel saving can be achieved by reducing overcooling and eliminating the unnecessary reheating process to reach comfort. As stated before, averting the overcooling would ultimately translate into saving energy, hence less fuel use and less emission.

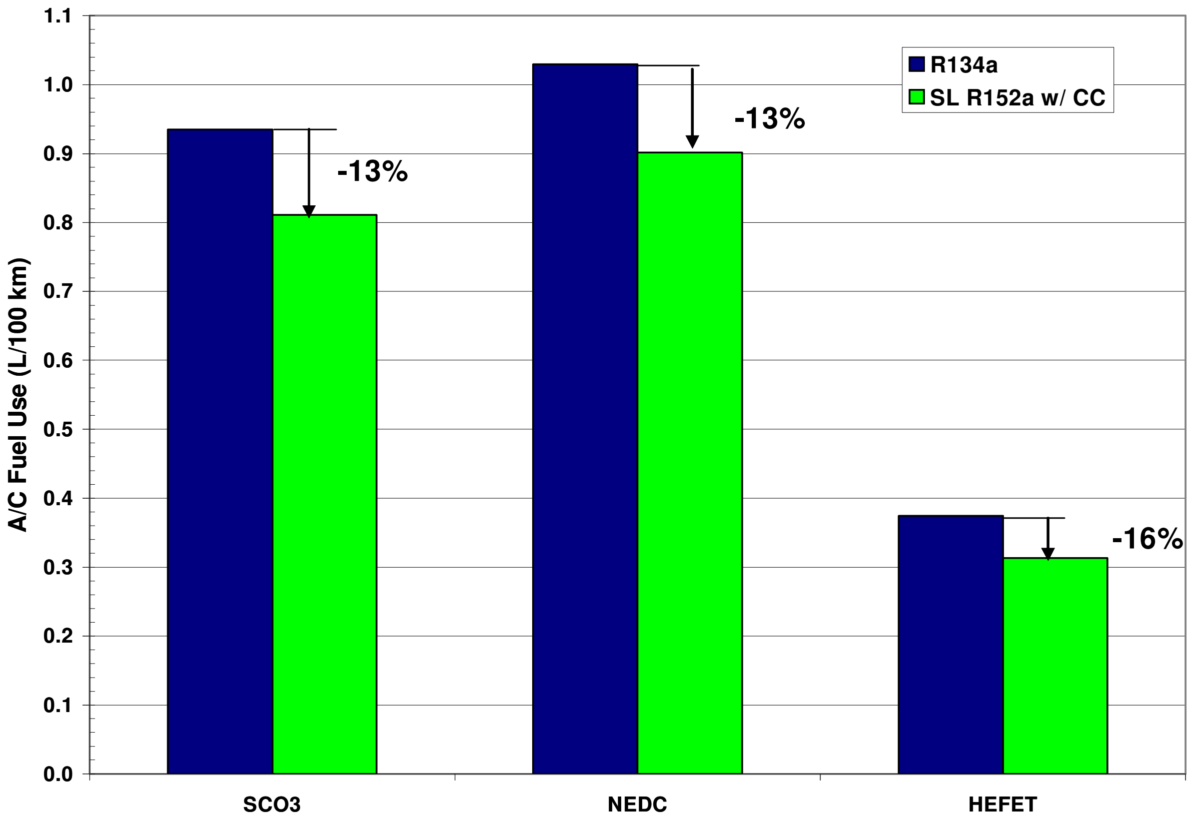


Figure 30 A/C Fuel Consumption

IX. SECONDARY LOOP SYSTEM DESIGN GUIDELINES

For comparison purpose, it is recommended to baseline the vehicle before equipping it with Secondary Loop system.

9.1 Overall System

- To meet maximum performance requirement, it is recommended to use higher displacement (10 – 15%)compressor

  Ex: Assuming same speed & isentropic efficiency for both compressors and using saturation properties of the refrigerants, @ 5, 10, 15 & 27 ^{0}C compressor in temperatures, the increase in displacement is around 11.5%

- Verify the desiccant type (XH-9 is required)
- Re-set engine cooling fan control to match R-152a properties

(recalibrated high side pressure transducer for R-152a. It is usually located around condenser)

- Check front end air flow (minimize air re-circulation)
- The TXV needs to be reset to match R-152a properties and the secondary loop applications
- Cooler coolant in temperature (instead of cooler air off) is recommended for freeze control
- Capacity controller
- Insulate refrigerant in line to chiller
- Insulate chiller
- Depending on the length, coolant lines could be insulated
- Coolant circuit:
  - Pump located at lowest point of circuit
  - Reservoir fill cap located at highest point (open to atmosphere for coolant circuit de-gassing)
  - Chiller on suction side of pump (pull coolant flow through chiller for better coolant distribution in the chiller)
  - Minimum reservoir size for faster cool down
  - Dual reservoir is recommended for idle - stop
  - Coolant mix: 32% Glycol / 70% water

More details on secondary loop system design guidelines are described in the design improvements section.

9.2 Potential Design Improvements

As with all design efforts, optimization is a series of trade-offs between various features or metrics. In the design of a Secondary Loop System, some key items to consider for optimization of both performance and minimum energy use of the design are:

- System Mass
- Heat Exchanger Effectiveness
- Component Package Size
- Compressor Efficiency
- Control Strategies for minimum Fuel Use
- “Cold” Storage for Energy Savings

Each of these features will be summarized in order to give options for performance and/or a reduction in the energy needed for operation.

9.2.1 System Mass

For the Astra demonstration vehicle, the additional mass added to support a Secondary loop system is summarized in table 3.

Table 3 System Mass

Component	**Added mass**	**Comment**
Chiller	1.8 kg (including fittings)	Based demonstration vehicle content
Cooler	Similar to evaporator	Based demonstration vehicle content
Pump	0.43 kg	
Coolant	3.2 kg	Function of project requirements
Total Mass	**5.45 kg**	Function of project requirements

As is shown, the chiller and coolant make up the majority of the added mass of the system. As an improvement, a reduction in mass can be achieved by optimizing the designs. Mass of the chiller depends on several factors: its required size to meet the heat transfer requirements in the vehicle, the necessary material gage to meet field reliability targets, its mounting features in the vehicle, its plumbing to interface with primary refrigerant and the coolant circuits. The amount of insulation added to the chiller assembly to minimize the environmental heat gain could add to the mass of the chiller assembly. Guidelines to select a chiller design based on these characteristics are:

1. Select the chiller size to meet the cooling requirements, as well as, balance the heat transfer and the chiller effectiveness versus other components in the system, particularly the pressure drop (coolant and refrigerant). An optimum chiller design will depend on the required heat exchanger surface area, the overall heat transfer coefficient, and the refrigerant / coolant flow rates.

2. The gage of the material selected must be consistent with the requirement of the SAE J639 specification, the material gage must meet the manufacturing, and durability needs as specified by each OEM.

3. Chiller mounting features such as brackets and fittings have an impact on the mass of the chiller. Directionally, it is preferred to encase the metallic heat exchanger in a plastic enclosure and use plastic features to mount the assembly to the vehicle.
4. The required plumbing to interface with the primary and secondary fluids will depend on the type of refrigerant control and the position of

the inlet and outlet fittings. The pass arrangement and the type of fittings used will influence the plumbing layout.

5. The addition of insulation to a chiller assembly is suggested to minimize heat loss; however, the environment where the chiller is located in the vehicle will determine the quality and the quantity of insulation.

The major source of additional mass is the amount of coolant used in the secondary fluid (glycol/water) loop. 0.75 gallon (3.2 liters) of a 30% glycol / 70% water mix was used in the Astra demonstration vehicle.

In any given application, there is a minimum amount of coolant required to fill the system volume while maintaining system operation. Additional coolant can be added to provide extra “cold” storage. For the Astra vehicle, the minimum amount was approximately 0.77 gallon (2.9 liters) was required to fill the system. As a reserve for minimum cold storage, an additional 0.08 gallon (0.3 liters) was added via an in-line reservoir. With this amount of coolant in the system, the Astra system was able to maintain vehicle comfort up to three times longer then the baseline R-134a production system. This level of performance was measured in the hot ambient conditions of Phoenix, AZ and it is shown in figure 31. The longer the desired A/C off time, the greater the coolant storage volume.

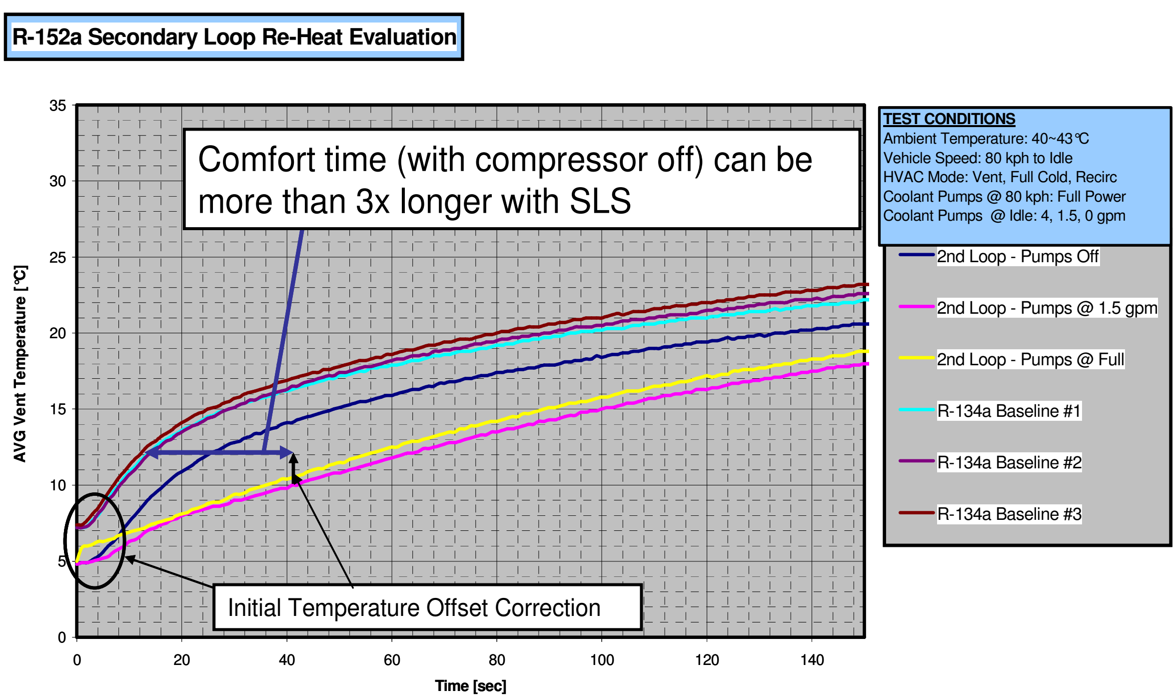


Figure 31 High Ambient System Warm-up with Cold Storage

9.2.2 Heat Exchanger Effectiveness

Although the secondary loop system has many benefits as we have discussed, the coolant circuit adds a "heat transfer resistance" that must be overcome to provide the necessary temperature difference. One common measure of how well heat is transferred in a given heat exchanger is its effectiveness. A heat exchanger effectiveness is defined as the ratio of the actual heat transfer over the theoretical maximum heat transfer in the heat exchanger. Since the coolant is a common fluid in the chiller and cooler, the two heat exchangers are considered coupled. London and Kays [9] have established an overall effectiveness of such coupled heat exchangers. Figure 32 shows the relationship between the power required by a secondary loop system as compared to an R-134a direct expansion system and the effectiveness of the heat exchangers at a fixed system load and range of evaporating and condensing temperatures. It is based on the Astra vehicle data with R-152a secondary loop system data.

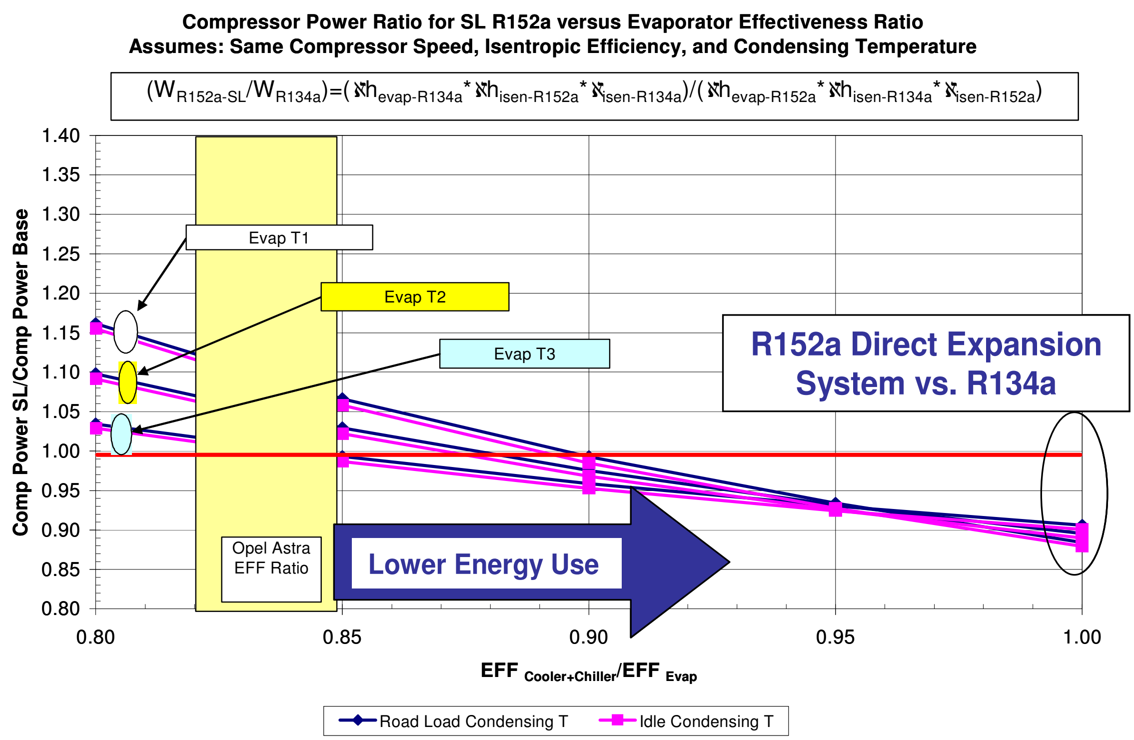


Figure 32 Compressor Power versus Effectiveness

It is important to note that by increasing the combined effectiveness of the cooler/chiller as compared to the direct expansion system evaporator effectiveness reduces the power required. The ratio for equivalent power is 0.9 or less. The fact that this ratio is less than 1 is a result of using the R-152a refrigerant. Therefore, as a guideline, designs that maximize the combined heat exchanger effectiveness will result in lower energy requirements for the secondary loop system.

9.2.3 Component Package Size

Packaging the components that make up a secondary loop system is one of the challenges in applying this technology to production vehicle applications. An advantage of the secondary loop system components is that they do not require a specific in-vehicle location to perform the required function. For example, power train cooling heat exchangers are located forward in the vehicle and require front-end airflow to operate effectively.

Guidelines to efficient packaging of the components:

- Locate the chiller away from significant heat sources such as turbo-charger units, exhaust manifolds, and piping.
- Add insulation to the chiller and coolant lines at a minimum to keep the packages as small as possible.
- Apply a brush-less motor driven pump for minimum package volume of the pump/motor assembly

9.2.4 Compressor Efficiency

The compressor has a major impact on system efficiency. As stated earlier, design modifications are recommended to improve efficiency and take advantage of R-152a and the secondary loop system attributes.

9.2.5 Control Strategy

Recently, technologies to reduce overcooling while maintaining passenger comfort and prevent fogging have been developed and introduced to the market. These technologies require extra content and added cost to the system in the form of externally controlled compressors and dedicated control systems.

In contrast, when using Delphi's Capacity Control technology with the secondary loop system, the benefit of reducing overcooling and consequently saving energy is an inherent part of the system. This Capacity Control) technology was applied to the secondary loop system Astra demonstration vehicle. The control capacity approach is inherent to the secondary loop system, only the coolant flow rate needs to be controlled and adjusted to achieve the desired level overcooling reduction. This control method can be integrated with an internally / externally controlled or fixed compressor.

Because of the control capacity system demonstrated on the Astra vehicle, when compared to the production manual A/C system, a minimum overcooling was implemented and resulted in a calculated reduction in fuel use of between 13% and

16% for various conditions. This estimates fuel use reduction is calculated using a model developed by the National Renewable Energy Laboratory (NREL) under the direction of John Rugh and is described earlier in section 8.3.

9.2.6 "Cold" Storage Systems

Another option that can be implemented in secondary loop system is the use of the stored cold fluid (Cold Storage) as a cold buffer to provide passenger comfort with the A/C compressor off. Two key transient conditions that can leverage the cold buffer to maintain comfort and reduced fuel use are:

- Idle-Stop or turning the engine off under idle conditions.
- Engine Management Systems (EMS) that can be adjusted to engage or disengage A/C compressor such that the energy use is reduced. In the case of small engine vehicles, drivability is improved with the compressor off. Added energy savings can be harvested with the compressor disengaged during accelerations and engaging it at maximum load to recover some of the braking energy (decelerations) to replenish the cold buffer. These EMS opportunities have been discussed with experts in engine control technology and they expressed that these viable opportunities need further study.

X. SUMMARY AND CONCLUSIONS

The goal of designing, building, and testing a commercially viable, energy-efficient, secondary loop HFC-152a mobile air conditioning system has been achieved. The system was applied to a 2007 Opel Astra to demonstrate this technology in a vehicle common in the European Union, wherein a phase-out of HFC-134a will begin in 2011.

- All component designs are based on known technologies with no anticipated barriers to timely production. The cooling performance of the system is equal or better to that of the baseline HFC-134a production system, as intended. The energy performance of the system is comparable to that of the baseline, due, in large part, to the inherent ability to easily control the cooling capacity of the system to provide comfort while avoiding the energy losses associated with overcooling, and subsequently reheating, the air. Compared to the production R-134a system, the energy use of the A/C compressor was shown through the NREL analysis to be 13% to 16% lower by applying Delphi's Capacity Control algorithm

- The Opel Astra was successfully demonstrated at the SAE Alternate Refrigerant Systems Symposium in July of 2007, during which attendees participated in an engineering review of the project and demonstration rides to assess the comfort of the system relative to other alternatives. Environmentally, by applying R-152a to a secondary loop system , Global

Warming impact of direct refrigerant emissions is reduced by at least 94% when compared to an equivalent R-134a system. This reduction takes into account both the lower GWP of R-152a and the lower mass of refrigerant required (40% less than the production R-134a system)

- The application of a secondary loop to vehicle air conditioning allows some unique fuel-saving opportunities. The secondary loop coolant system acts as a cold buffer that can provide cooling for a period after the compressor is disengaged. By combining the control of this cold buffer with the engine management system, the compressor can be engaged or disengaged when the opportunity exists for fuel savings. For example, the compressor can be disengaged during vehicle accelerations save energy and be fully engaged during vehicle decelerations to use vehicle engine braking energy to replenish the cooling capacity of the cold buffer. Idle-stop, i.e. turning off the engine, and thereby the A/C compressor when the vehicle is stooped for brief periods can be enhanced by the ability of the cold coolant buffer to provide comfort under compressor off conditions as compared to a direct system.

Application of a secondary loop to vehicle A/C is a new concept that carries with it its own challenges with respect to heat exchange and system controls. Accordingly, as with any new system, many opportunities exist for improvements. Some of these, such as control technologies are highlighted in this report.

ACKNOWLEDGMENT

The authors would like to express their sincere thanks to Dr. Stephen O. Andersen and Kristen Taddonio of the U.S. Environmental Protection Agency for their support, as well as John Rugh of the US DOE National Renewable Energy Laboratory for his modeling, analysis and collaboration on the vehicle Fuel Use. Thanks also to General Motors for the kind assistance in supplying the vehicle for the project. The authors also whish to thank the Delphi Secondary Loop Team for the huge effort in seeing the project through. Thanks also to the Peer review team for inputs, comments, and support.

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